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ABSTRACT

This is the second of two training courses designed to develop the capability of practitioners in the home building industry to design solar heating and cooling systems. The course is organized in 23 modules to separate selected topics and to facilitate learning. Although a compact schedule of one week is shown, a variety of formats can be arranged. In general, the course progresses from simple sizing procedures for making preliminary estimates of collector area requirements, to a computer-aided method, and finally to automated design techniques. Such details as system economics, energy conservation trade-offs, and component selections are also presented. (Author/MLF)

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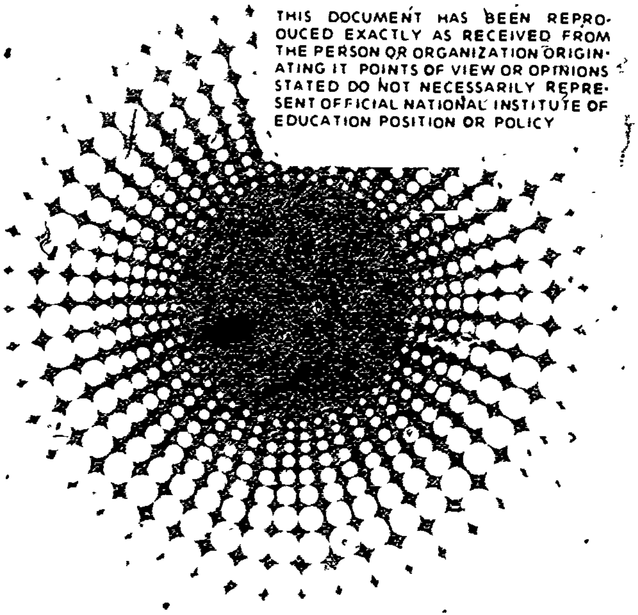

SOLAR HEATING AND COOLING OF RESIDENTIAL BUILDINGS
DESIGN OF SYSTEMS

Prepared by
SOLAR ENERGY APPLICATIONS LABORATORY
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SOLAR ENERGY APPLICATIONS LABORATORY
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PREFACE

The primary purpose of this training course is to develop the capability of practitioners in the Home Building industry to Design Solar Heating and Cooling Systems for Residential Buildings. The goal is to have this course implemented nationwide to train practitioners in the requisite skills to integrate solar energy systems into residential buildings.

Recent estimates indicate that a substantial amount of domestic space and water heating in the United States will be accomplished by solar energy in the near future. However, significant implementation can only be achieved if substantial capabilities are created among the professions and trades in the building industry to install solar systems.

This training course, and a companion course titled Sizing, Installation and Operation of Systems for Solar Heating and Cooling of Residential Buildings, are courses to train home designers and builders in the fundamentals of solar hydronic and air systems for space heating and cooling and domestic hot water heating for residential buildings. The modularized structure of the training courses provides considerable latitude in organization and presentation, especially with regard to the time period over which the course could be presented. At Colorado State University, the course is presented in five continuous days, but a longer period of time utilizing only evening hours could be used just as effectively. The structure also provides for verification that participants have achieved anticipated levels of understanding. At CSU, validation is in the form of daily evaluations by the participants especially with regard to material content and methods of presentation. The instructors interact and respond to the evaluations and alter their methods of presentation to meet the needs of particular groups of trainees.

COURSE DEVELOPMENT

This training course was developed by the staff of the Solar Energy Applications Laboratory and vocational education specialists at Colorado State University in cooperation with the NAHB Research Foundation, Inc., Rockville, Maryland. A national advisory committee was established to provide advice and general guidance to the project staff regarding direction and content of the training courses. The committee members were from various sectors of the home-building industry, and also teachers, architects, engineers and representatives from governmental agencies.

In determining curriculum content, a rigorous procedure was followed to develop course standards and needs by interacting with architects, engineers, building contractors and installers of heating, ventilating and air conditioning systems in residential buildings. From the standards and needs, objectives for the course were developed and the curricular materials were then prepared.

ABOUT THE AUTHORS

This manual for the training course was prepared with the cooperative efforts of many people under the organizational efforts of C. Byron Winn. The program for development of both the Design and the Installation courses was directed by Susumu Karaki with George O. G. Löf as senior advisor. As the following short biographical sketches indicate, the authors of this manual have, individually and collectively, considerable experience in the design, installation, operation and maintenance of solar systems for space heating and cooling and domestic hot water heating.

C. Byron Winn -- Dr. C. Byron Winn is Professor of Mechanical Engineering at Colorado State University and has been actively involved in solar heating and cooling systems since 1973. He has published widely and has designed and installed both air-heating and liquid-heating solar systems in seven residences in various locations in Colorado.

Byron Winn organized and directed the development of the course and prepared a major portion of this manual. In addition to his experience as a designer and installer of systems he has conducted research and taught courses in solar energy applications.

Susumu Karaki -- Dr. Karaki has been a member of the faculty at Colorado State University for the past 19 years. He is Associate Director of the Solar Energy Applications Laboratory and Professor of Civil Engineering. Being involved in solar energy research since 1973, he has directed a number of research projects in solar energy utilization. Susumu Karaki has served on several committees of the International Solar Energy Society, and has been a member of U. S. teams for international information exchange on solar energy utilization.

George O. G. Löf -- Dr. Löf has specialized in solar energy utilization for over thirty years and pioneered in the development of solar heating and cooling systems. As Director of the Solar Energy Applications Laboratory he is responsible for the considerable progress made in development of solar systems at Colorado State University and elsewhere. His accomplishments have earned him worldwide recognition and among his many awards is the Lyndon Baines Johnson award for outstanding service to society.

Gearold R. Johnson -- Dr. Johnson is Associate Professor of Mechanical Engineering at Colorado State University. In addition to his interests in solar energy applications, he has been actively engaged in energy conservation studies and the application of microprocessors for management of energy utilization in structures.

Dan S. Ward -- Dr. Ward joined the staff of the Solar Energy Applications Laboratory at Colorado State University in 1973 and presently serves as Assistant Director of the Laboratory as well as Assistant Professor of Civil Engineering and Physics.

Dan Ward has considerable experience with solar heating and cooling systems including supervision of design, construction and installation of the solar systems in three Solar Houses at Colorado State University. He is serving as chairman of the ASTM subcommittee on development of testing standards for solar energy systems.

William S. Duff -- Dr. Duff is Associate Professor of Mechanical Engineering at Colorado State University. He has expertise in system optimization and applies his interests to solar heating and cooling systems as well as to other applications for solar energy.

Sanford B. Thayer -- Dr. Thayer is Associate Professor of Mechanical Engineering. He has had extensive experience in engineering economics which is applied to solar heating systems in this manual.

Milton E. Larson -- For the past 25 years, Dr. Larson has been engaged in educational work, with technical education and trade and industrial education as the focus of activity. He is Professor of Vocational Education at Colorado State University and has served as head teacher-trainer for technical education in the Department of Vocational Education for the last eleven years.

Milton Larson along with Dr. Valentine provided expert advice to the staff in developing the training course and this manual.

Ivan E. Valentine -- Dr. Valentine, along with Dr. Larson served the staff who prepared this manual as a vocational education specialist. He has considerable experience in curriculum development in all areas of technical education and has, additionally, practical experience as a consulting engineer, and heating and plumbing contractor.

Ivan Valentine's extensive experience in vocational technical education contributed significantly to the development of this practical training course and manual.

NAHB RESEARCH FOUNDATION, INC.

The NAHB Research Foundation through Ralph J. Johnson, Staff Vice President and Director, and H. W. Anderson participated in the development of the training course and this manual through critical reviews and many helpful comments. Mr. Johnson in particular carries with him over 30 years experience in housing and home-building research, and in housing for nearly twenty years.

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The staff is especially indebted to Wendy Asa and Kathi McKenna for preparing and organizing the manuscripts for this manual. Their patience and service are truly appreciated.

NOTICE

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AUTHORS OF MODULES IN THIS MANUAL

MODULE	TITLE	AUTHOR(S)
1	Course Orientation	S. Karaki
2	General Descriptions of Solar Heating and Cooling Systems	C. B. Winn
3	Solar Radiation Information for Design Purposes	C. B. Winn & S. Karaki
4	System Design Guidelines	C. B. Winn & S. Huck
5	Heating and Cooling Load Analyses	S. Karaki & G. Johnson
6	Simplified Design Calculations	C. B. Winn
7	Detailed Design Methods	C. B. Winn
8	Economic Considerations	S. Thayer & C. B. Winn
9	Energy Conservation Trade-offs	G. Johnson
10	Detailed Design Calculations	C. B. Winn
11	Collectors	G. O. G. Löf
12	Storage Systems	W. S. Duff
13	Laboratory	C. B. Winn
14	Computer-Aided F-Chart Calculations	C. B. Winn
15	System Controls	C. B. Winn
16	Selection of Subsystem Components	C. B. Winn
17	Solar Cooling Systems	S. Karaki
18	Automated Design Techniques	C. B. Winn
19	Service Hot Water Systems	G. O. G. Löf
20	Design Case Study	C. B. Winn
21	Structural, Mechanical and Scheduling Considerations	S. Karaki
22	Future Prospects for Solar Heating and Cooling Systems	D. S. Ward
23	Buyer's Guide	G. O. G. Löf

DESIGN OF SOLAR HEATING AND COOLING SYSTEMS

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THE PRACTICAL ASPECTS OF
DESIGN OF SOLAR HEATING AND COOLING SYSTEMS
FOR
RESIDENTIAL BUILDINGS

MODULE 1

COURSE ORIENTATION

SOLAR ENERGY APPLICATIONS LABORATORY
COLORADO STATE UNIVERSITY,
FORT COLLINS, COLORADO

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INTRODUCTION

The Solar Energy Applications Laboratory at Colorado State University in cooperation with the NAHB Research Foundation has developed two practical training courses for the design, installation and operation of solar heating and cooling systems for residential buildings. This course is titled DESIGN OF SYSTEMS and the other is SIZING, INSTALLATION AND OPERATION OF SYSTEMS.

The use of solar energy to provide the comfort conditions in residential buildings and serve the hot water needs is a practical realization for many parts of the country where costs of electricity and fossil fuels are steadily increasing. Although there is considerable interest in solar systems throughout the country, the numbers of solar houses are relatively few, largely because there is a serious lack of qualified personnel to apply the technology. Substantial capabilities are needed among the professions and trades involved in the building industry to design and install solar systems if widespread application is to take place in this country.

This training course was prepared to develop practical skills to design solar heating and cooling systems. Over a period of one week, the course provides 44 hours of instruction, practice with computations and detailed inspections of working systems.

OBJECTIVES

The objectives of the training course are to develop capabilities in the trainee to:

1. Design solar heating and cooling systems for residential buildings, and make performance estimates and economic analyses.

2. Advise clients on particular types of systems best suited for their needs.
3. Plan and supervise construction of buildings which include solar systems.
4. Explain basic operating characteristics of solar heating and cooling systems to others.

SCOPE

This course is limited in scope to the design of solar heating and cooling systems for residential buildings, with primary emphasis on heating systems. Although solar cooling systems are discussed, design and economic analyses of systems for only solar cooling of buildings are not included, principally because such systems are not, as yet, economically competitive with standard refrigeration systems. However, where solar heating systems can be economically justified, adding solar-operated cooling units to form integrated solar systems may be possible. Designs of such integrated systems are included in this course.

Although the basic design principles apply to any solar heating and cooling system, the user is advised that many of the design charts in this manual apply only to residential buildings. When solar systems are considered for office, commercial or industrial buildings, the application may be sufficiently different that alternative procedures should be followed.

COURSE ORGANIZATION

The course is organized in modules to separate selected topics and facilitate learning. Although a compact schedule of one week is shown in Figure 1-1, variety of formats can be arranged.

	SUNDAY	MONDAY	TUESDAY	WEDNESDAY	THURSDAY	FRIDAY
0800		MODULE 1 (30 min) Course Orientation MODULE 2 (90 min) General Descriptions of Solar Heating and Cooling Systems	REVIEW AND QUESTIONS (30 min) MODULE 7 (90 min) Detailed Design Methods	REVIEW AND QUESTIONS (30 min) MODULE 11 (90 min) Collectors	REVIEW AND QUESTIONS (30 min) MODULE 15 (90 min) System Controls	REVIEW AND QUESTIONS (30 min) MODULE 20 (90 min) Design Case Study
1000		COFFEE (30 min)	COFFEE (30 min)	COFFEE (30 min)	COFFEE (30 min)	COFFEE (30 min)
		MODULE 3 (90 min) Solar Radiation Information for Design Purposes	MODULE 7 cont (30 min) Detailed Design Methods MODULE 8 (60 min) Economic Considerations	MODULE 11 cont (30 min) Collectors MODULE 12 (60 min) Storage Systems	MODULE 16 (90 min) Selection of Subsystem Components	MODULE 21 (45 min) Structural, Mechanical and Scheduling Considerations MODULE 22 (45 min) Future Prospects for Solar Heating and Cooling Systems
1200		LUNCH (60 min)	LUNCH (60 min)	LUNCH (60 min)	LUNCH (60 min)	LUNCH (60 min)
1300	1400 - Registration 1445 - Solar House Tours	MODULE 4 (60 min) System Design Guidelines MODULE 5 (60 Min) Heating and Cooling Load Analyses	MODULE 9 (120 min) Energy Conservation Trade-offs	MODULE 13 (120 min) Laboratory	MODULE 17 (120 min) Solar Cooling Systems	MODULE 23 (60 min) Buyer's Guide Review and Summary (60 min)
1500		COFFEE (30 min)	COFFEE (30 min)	COFFEE (30 min)	COFFEE (30 min)	COFFEE (30 min)
		MODULE 6 (90 min) Simplified Design Calculations	MODULE 10 (90 min) Detailed Design Calculations	MODULE 14 (90 min) Computer-Aided F-Chart Calculations	MODULE 18 (45 min) Automated Design Techniques MODULE 19 (45 min) Service Hot Water Systems	Evaluation of the Course by participants
1700		ADJOURN	ADJOURN	ADJOURN	ADJOURN	ADJOURN
1730	Reception and Dinner					Dinner and Awards

Figure 1-1. Course Schedule

In general, the course progresses from simple sizing procedures for making preliminary estimates of collector area requirements, to a computer-aided method and finally to automated design techniques. Such details as system economics, energy conservation trade-offs and component selections are also presented.

Opportunities for review are provided each day in the schedule of Figure 1-1, and participants are encouraged to use the time to clarify any difficulties encountered. At the end of each day, the trainees are requested to evaluate the course materials and methods of presentation. These evaluations will assist the instructors to respond to the needs of the particular group of trainees in the course.

SYNOPSIS OF COURSE CONTENT

TOUR OF SOLAR HOUSES

A pre-course tour of solar houses in the local area is provided to give trainees an opportunity to see different styles of homes which have different solar systems. The systems are briefly described and performance details of systems are given when such information is available. The duration of the tour is about 3 hours. After the tour there is an informal reception and dinner where the instructors and trainees are introduced. A post-dinner talk on the energy problem is presented.

MODULE 1. COURSE ORIENTATION

The objective of the training course is to develop capabilities among practitioners in the home-building industry and other interested persons to design solar heating and cooling systems that will provide a major portion of the annual heating needs of a residential building. In this introductory module, a summary of the contents of the course is presented and a schedule for the course is given.

A brief preview is provided to highlight the course so that trainees can appreciate the relationship of each module to the entire course. The course progresses gradually from basic concepts and preliminary design procedures toward more complex aspects and detailed calculations.

MODULE 2. GENERAL DESCRIPTIONS OF SOLAR HEATING AND COOLING SYSTEMS

Several operating solar houses are described in the module along with descriptions of the houses, the types of systems, and design data for the systems. The purpose of this module is to establish a base of reference for the types of residential solar systems that will be described in greater detail in the course. Basic arrangements of liquid-heating and air-heating solar systems that are practical to install are described.

From the descriptions of the systems the trainees can better appreciate the relationships of collector areas to floor areas and volumes of heat storage to collector areas. Although costs are likely to vary somewhat in the future, the range of costs for solar systems is also given. The reader is cautioned that some of the systems are experimental units, where extra features have been designed into the system to permit alternative modes of operation. Such systems are likely to be more expensive than systems that would be installed in normal residential buildings.

Schematic diagrams are used to describe basic arrangements of solar systems and to trace the flow of heat from collectors to storage and from storage to the building space. Solar heated domestic hot water systems are also introduced in the module.

MODULE 3. SOLAR RADIATION INFORMATION FOR DESIGN PURPOSES

Understanding the intensity of solar radiation at the surface of the earth is basic to the design of solar systems. Regardless of the type of collector used, there is a limit to the amount of heat that can be obtained from a unit area of the collector. There is considerable variation in the amount of heat that can be collected from the sun with a given collector area, depending upon the tilt and orientation of the collectors. From an economic point of view, both the collector tilt and orientation should be set for maximum collection for the season. If the system is designed only for heating purposes, the collector should be arranged for low sun angles. On the other hand, if the system is to heat and cool the building, a flatter tilt would be more suitable to maximize energy collection during the entire year.

In this module, the trainees are taught how to calculate the solar radiation intensity on a tilted collector surface. Complex equations are given in the text, but all calculations can be made by use of charts and tables which involve only additions and multiplications of numbers.

MODULE 4. SYSTEM DESIGN GUIDELINES

A system designer may often be faced with the task of making a quick estimate of the collector area (and cost) required for a solar system in a given building. If he is serving a client and first cost is of primary concern to the owner, the designer usually cannot afford the time to make detailed calculations before a decision is made by the client. On the other hand, it is important to be able to make a quick estimate of the fraction of the annual heating load that can be provided by a system with a given collector area.

Approximate methods for sizing collectors are described in the module, which involve the solar radiation on a tilted collector, the January heating load for the building and a pre-selected fraction of the annual heating load which the system is to supply. From the collector area, the heat storage volume and the fluid flow rates are determined. The methods apply to both air-heating and liquid-heating systems which have "standard" arrangements of components.

MODULE 5. HEATING AND COOLING LOAD ANALYSES

The objective of this module is to present methods for calculating the heating and cooling loads of a residential building. The procedures described in the manual are essentially those in the ASHRAE Handbook of Fundamentals, 1972.

For sizing furnaces and boilers for non-solar residential buildings, heating load calculations have not needed to be precise because the costs of heating units are minor relative to building costs. If a 100,000 Btuh furnace is used where a 60,000 Btuh unit is needed, the added cost and loss of furnace efficiency in the larger unit have not been significant issues because fuel has been plentiful and cheap.

In designing a solar system, a design heat loss rate from a building is needed to size the auxiliary unit, and monthly average building heating loads (determined from the design heat loss rate) are needed to determine an economical system size. Over-sized or under-sized solar systems are uneconomical, and therefore, a reliable estimate of the building heating load is needed.

MODULE 6. SIMPLIFIED DESIGN CALCULATIONS

The trainees are given an opportunity to design a solar system using the guidelines presented in Module 4. A common problem is assigned to the entire class, and those who complete the class problem can design a system for their locality.

MODULE 7. DETAILED DESIGN METHODS

The simplified design methods presented in Module 4 are usually not adequate for final design purposes because only typical systems are represented, and design parameters that characterize specific collectors are not included in the method. When systems are designed with specific collectors, and efficiency curves are available for those collectors, the designs can be improved, and better predictions of useful solar energy collection can be made.

The method described in this module is the procedure developed by Duffie, Beckman and Klein at the University of Wisconsin and is called the f-chart design procedure. By specifying the details of the system used to collect solar energy and by specifying the end use, the method can be used to predict the performance of systems which provide energy for space heating and domestic hot water heating. Currently, only the performance of the systems using flat-plate collectors can be obtained with f-charts.

MODULE 8. ECONOMIC CONSIDERATIONS

This module presents methodology that may be used to determine the economics of solar heating systems. One method is to make a break-even analysis to predict the price of fuel when the annual cost for a non-solar system will equal the annual cost for a solar system. Although this methodology does not give a true economic picture of solar systems, it is a method used by some in a decisionmaking process. The method of analysis is explained using natural gas as the comparative fuel energy with solar, but the method is equally applicable to compare other energy forms such as fuel oil, propane and electricity with solar energy.

A more realistic economic analysis should include inflation rates, operating and maintenance costs, property taxes, insurance and credit for taxes as well as mortgage payments and fuel costs. A method for comparing the annual cash flows for non-solar as well as solar systems is presented in the module.

MODULE 9. ENERGY CONSERVATION TRADE-OFFS

In recent years, home designs are including measures to reduce the quantity of heating (and cooling) needed to maintain comfort conditions. Much can be accomplished with architectural treatments of the building exterior, with regards to shape of the building, orientation, fenestration, and air locks, but other energy-conserving measures such as insulation, storm windows and doors, and internal temperature settings can be effective in reducing the energy needs in the building.

The cost-effectiveness of several energy conserving measures are discussed in the module using a basic house design and comparing the reduction in energy requirements with several conserving measures. The cost for effecting energy conservation is compared with the savings in energy consumption.

MODULE 10. DETAILED DESIGN CALCULATIONS

The trainees are given an opportunity to make detailed design calculations using the f-chart procedure described in Module 7. Four problems are assigned in a progressive order of complexity to familiarize participants with the calculation procedure.

MODULE 11. COLLECTORS

Descriptive and specific details of solar collectors are described in the module. There are two basic types of collectors, flat-plate and concentrating. Concentrating collectors focus beam radiation from the sun onto small absorber surfaces and develop high temperatures in heat transfer fluids. Flat-plate collectors have no focusing, and the total radiation from the sun and reflections from other surfaces are used to heat fluids that are in contact with the absorber. Flat-plate collectors are the only types described in detail in this module, and general performance curves for a number of collectors are given.

MODULE 12. STORAGE SYSTEMS

The purpose of thermal storage in solar heating and cooling systems is to provide heat for use during non-sunshine hours. In practical systems, heat must be easily storable and readily reclaimable for use in the building. Sensible heat in water or rocks is the most common way to store heat. Phase-change material permits large quantities of heat to be stored in a small amount of mass and is, therefore, a possible way to reduce the volume of storage needed for residential solar systems as compared with a water tank or rock bin. More research is needed at this time to develop the use of the volume required for a phase-change material, before such material will become practical to use.

Details of water storage and rock bins are described in the module, as well as other performance and design characteristics of storage units in solar heating and cooling systems.

MODULE 13. LABORATORY

The laboratory session is an opportunity for trainees to inspect working systems carefully and to learn where to take temperature measurements and place control sensors in working systems and model systems.

MODULE 14. COMPUTER-AIDED F-CHART CALCULATIONS

Having made design calculations with a hand calculator using the f-chart method, the trainees are instructed on the use of a computer-aided interactive f-chart design procedure. The interactive computer program enables a number of iterative designs to be made quickly to reach an economical solution.

MODULE 15. SYSTEM CONTROLS

The basic function of controls in solar heating and cooling systems is to switch on pumps and blowers and operate valves and dampers in response to the heating or cooling needs inside the building and to the available sunshine (or lack of sunshine) on the collectors. As the occupant of the building needs to be concerned only with the thermostat setting, the entire system is controlled automatically. The control logic and types of available controls are explained in this module.

MODULE 16. SELECTION OF SUBSYSTEM COMPONENTS

In addition to collectors and storage units, the selection of heat exchangers, pumps and blowers and valves and dampers is important to system performance. The type of heat exchanger in a liquid system can materially affect the collector operating temperature and system efficiency. Similarly, the pump or blower capacity can affect the fluid temperature rise through a collector. The selection of subcomponents of a system is explained in this module.

MODULE 17. SOLAR COOLING SYSTEMS

Solar cooling units that are integrated with solar heating systems are discussed in the module. The only commercially available solar cooling unit (1977) is a lithium-bromide-water absorption machine, but other possible units are described. Evaporative cooling, although it does not depend upon solar energy, can be integrated with an air-heating system, utilizing the rock bed to store "cool". That is, the rock bed is cooled down at night and used to cool the warm room air during the day. This method, however, is limited for use in arid and semi-arid regions of the country.

MODULE 18. AUTOMATED DESIGN TECHNIQUES

The design techniques presented in Modules 4 and 7 are based upon "standard" system arrangements. When non-standard arrangements are to be designed, with high performance collectors or heat pumps, for example, the previous methods will not provide performance predictions for such systems, and computer simulations should be used. There are at least three programs available, TRNSYS, SIMSHAC, and SOLCOST, and the latter is an automated program that can be used even if the user is not familiar with computer techniques. The utility of these programs is discussed in this module.

MODULE 19. SERVICE HOT WATER SYSTEMS

A solar hot water heating system can be used for domestic service. The two major types of solar water heaters are circulating and non-circulating, with several variations of each type. In its simplest form, a solar water heater consists of a flat-plate collector with an insulated tank with water circulated through the collector by thermosyphon action. More complex systems involving pumps, antifreeze solutions with heat exchangers, and drain-down systems are described.

MODULE 20. DESIGN CASE STUDY

A design of a system using the automated design procedure is illustrated for the class of trainees.

MODULE 21. STRUCTURAL, MECHANICAL AND SCHEDULING CONSIDERATIONS

The scheduling of a solar installation, especially for new buildings, can be arranged with concurrent and sequential activities to minimize installation problems. Such items as installation of the storage unit before floor joists are placed, and structural considerations to support the weight of tanks and rock boxes, are described. Mechanical considerations and arrangements of other components are also described.

MODULE 22. FUTURE PROSPECTS FOR SOLAR HEATING AND COOLING SYSTEMS

New designs for the components of solar systems are undergoing research, development and testing. Collectors that are much improved in performance, and possibly direct contact heat exchangers for liquid systems that combine with storage, may become technically and economically advantageous for use in the future. Prospects for improvements in solar systems look bright, and when improvements are proven to be practical they should be considered for use.

MODULE 23. BUYER'S GUIDE

To select proper equipment for solar systems, the buyer should have knowledge of standards, equipment warranties, and performance of components in a system. He should have an understanding of the performance of liquid-heating and air-heating systems and their respective advantages and disadvantages so that rational choices can be made for use in specific buildings. Considerations for costs of systems, reliability, operating costs and maintenance requirements can also guide buyers.

TRAINING COURSE IN
THE PRACTICAL ASPECTS OF
DESIGN OF SOLAR HEATING AND COOLING SYSTEMS
FOR
RESIDENTIAL BUILDINGS

MODULE 2

GENERAL DESCRIPTIONS OF SOLAR HEATING AND COOLING SYSTEMS

SOLAR ENERGY APPLICATIONS LABORATORY
COLORADO STATE UNIVERSITY
FORT COLLINS, COLORADO

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OBJECTIVES OF THIS MODULE

TRAINEE-ORIENTED OBJECTIVE

From the material presented in this module, the trainee should become familiar with some design data for several existing solar heating and cooling systems.

SUB-OBJECTIVES

1. To develop an understanding of solar system costs.
2. To develop an understanding of solar system sizing considerations.

Several existing systems will be described in this module. In addition, general descriptions of both air and liquid systems will be presented.

REPRESENTATIVE EXISTING SYSTEMS

SOLAR VILLAGE AT CSU

Figure 2-1 shows the Solar Village on the Foothills Campus at Colorado State University. The building on the left, CSU Solar I, completed in July 1974, is the world's first solar heated and cooled residential type structure. The solar system consists of 768 square feet of flat-plate collectors with two glass covers, 1100-gallon water storage tank, a 3-ton lithium bromide absorption chiller, auxiliary fuel-fired boiler, and associated pumps, valves and controls. A solution of ethylene glycol and water is used as the transport medium in the collectors. The storage medium is water. Some of the design characteristics are listed in Table 2-1. The interesting features to note from Table 2-I are the

Figure 2-1. Environmental Village at Colorado State University

ratio of collector area to total heated and cooled floor space, 1 to 4, the ratio of storage volume to collector size, 1.5 gallons for each square foot, and the ratio of system cost to collector size, approximately $13 per square foot.

The house in the center on Figure 2-1 is CSU Solar II. The solar system consists of 736 square feet of air-heating solar collectors, pebble bed storage, an auxiliary fuel-fired furnace and an evaporative cooling unit. This solar system utilizes air as the transport medium in the collectors and a pebble bed heating and cooling storage unit. One of the principal objectives in constructing these two houses was to obtain comparisons between the performances of two different operating systems in nearly identical structures when subjected to almost identical climatological conditions. Some of the design data for CSU Solar II are listed in Table 2-2. We note that the ratio of collector area to floor space is approximately 1 to 4, the ratio of storage volume to collector area is approximately 0.5 cubic feet of rocks per square foot of collector, and the ratio of system cost to collector size is approximately $22 per square foot of installed collector. The cost of the air system in Solar House II appears to be over twice that of the liquid system of Solar House I. However, these systems were constructed in a different manner, and accountings of actual costs are, therefore, different. The system in Solar House I was constructed with student assistance under faculty direction, and the system in Solar House II was installed by a commercial installer with supervision by the supplier of the solar system hardware. As we shall see later, the costs for system installation in Solar House II are more representative of actual 1976 costs.

Table 2-1. Design Data for CSU Solar House I

Floor Space	$\sim$3000 Ft^2
Collector Area	$\sim$ 768 Ft^2
Collector Type	Flat-Plate Liquid
Transport Medium	Water and Ethylene Glycol
Storage Medium	Water
Storage Size	$\sim$1100 Gallons
System Cost	\$10,100 (estimate)

Table 2-2. Design Data for CSU Solar House II

Floor Space	$\sim$3000 Ft^2
Collector Size	$\sim$ 736 Ft^2
Collector Type	Flat-Plate Air
Storage Volume	363 Ft^3
Storage Medium	Pebbles ($\sim$20 tons)
System Cost	$\sim$\$17,000

The house on the right in Figure 2-1 is CSU Solar III. The system consists of evacuated tube collectors, a 1200-gallon water storage tank, a 2-ton lithium bromide absorption chiller, an electrical auxiliary heater and associated pumps, valves and controls. As with CSU Solar I and II, Solar House III is both heated and cooled. Some of the design data for CSU Solar III are listed in Table 2-3. We note that the ratio of collector area to floor space is approximately 1 to 4, the ratio of storage volume to collector area is approximately 2.5 gallons of water per square foot of collector, and the ratio of system cost to collector size is approximately $55 per square foot of installed collectors.

OTHER SYSTEMS

Figure 2-2 shows an artist's rendering of the first completely private solar heated residential type structure that was built in the Rocky Mountain Region. No funds were made available from any governmental organization for the purpose of designing or constructing this solar heated house, and it was not part of any demonstration program. Rather, it simply involved a private citizen who was interested in putting a solar heating system into his house. He was motivated largely by the difficulties he had experienced in obtaining propane for space heating purposes. The system consists of flat-plate air-heating collectors, pebble bed storage and an electric auxiliary heater. The construction was completed in the fall of 1974. Design data for the house are shown in Table 2-4. Here we see that the ratio of collector area to floor area is 1 to 3, the ratio of storage volume to collector area is approximately .25 cubic feet of rock per square foot of collector, and the system cost is approximately $10 per square foot of installed

Table 2-3. Design Data for CSU Solar House III

Floor Space	~3000 Ft^2
Collector Size	512 Ft^2
Collector Type	Evacuated Tube (Liquid)
Storage Volume	1200 Gallons
Storage Medium	Water
System Cost	$28,000 (estimate)

Figure 2-2. The Wellington Residence

Table 2-4. Design Data for Wellington Residence

Floor Space	~3600 Ft^2
Collector Size	~ 1200 Ft^2
Collector Type	Flat Plate (Air)
Storage Volume	336 Ft^3
Storage Medium	Pebbles (~19 tons)
System Cost	$13000

collector. This was one of the earliest commercial installations installed under contract.

Figure 2-3 shows the Eco-Era House No. 1. This was the first solar spec house built in the United States. The system, completed in June 1975, is an air-heating solar system, and the total construction, including the solar system, was privately funded. A modified version of this house has been built in Fort Collins as part of the U. S. Department of Housing and Urban Development demonstration program. Design data for Eco-Era No. 1 are shown in Table 2-5. Here we see the ratio of collector size to floor area is approximately 1 to 5, the ratio of storage volume to collector size is slightly less than .5 cubic feet of rocks per square foot of collector, and the system cost was approximately $18.50 per square foot of installed collector.

Figure 2-4 shows the Eco-Era No. 2 house that was completed in September 1976. Design data for Eco-Era No. 2 are shown in Table 2-6. We observe that the ratio of collector size to floor area is 1 to 5, the ratio of storage volume to collector size is approximately 0.5 Ft^3 per square foot of collector, and the system cost was approximately $28 per square foot of installed collector. The Eco-Era 1 and 2 houses are of particular interest since they represent multiple experiences at building similar houses using the same basic solar system.

Figure 2-5 shows the first solar heating system on a military installation. This was a retrofit of one of the housing units at the United States Air Force Academy. It included both a roof array and a ground array of flat-plate water heating collectors. Design data for this solar installation are shown in Table 2-7. In this case,

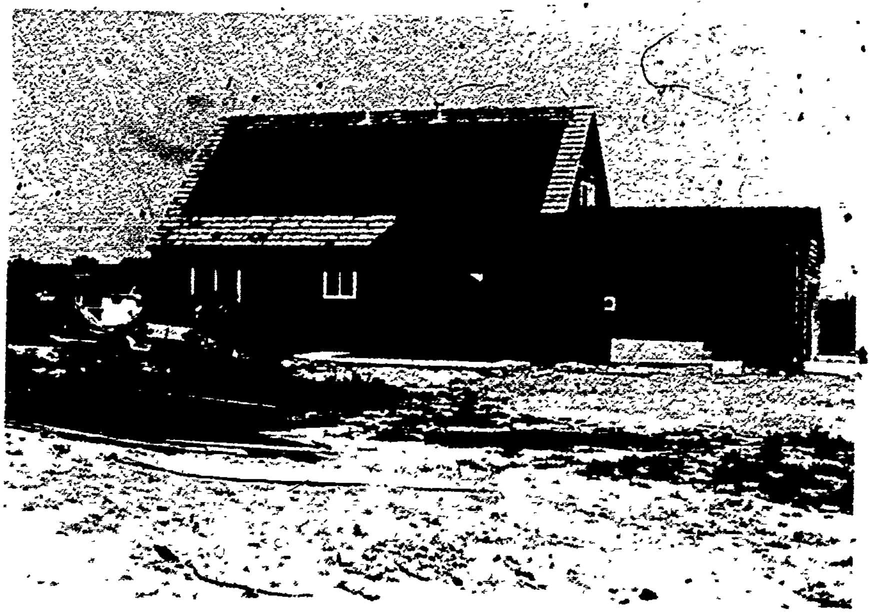

Figure 2-3. Eco-Era No. 1

Table 2-5. Design Data for Eco-Era No. 1

Floor Space	1560 Ft^2
Collector Size	432 Ft^2
Collector Type	Flat Plate (Air)
Storage Volume	180 Ft^3
Storage Medium	Pebbles (~10 tons)
System Cost	$8,000

Figure 2-4. Eco-Era No. 2

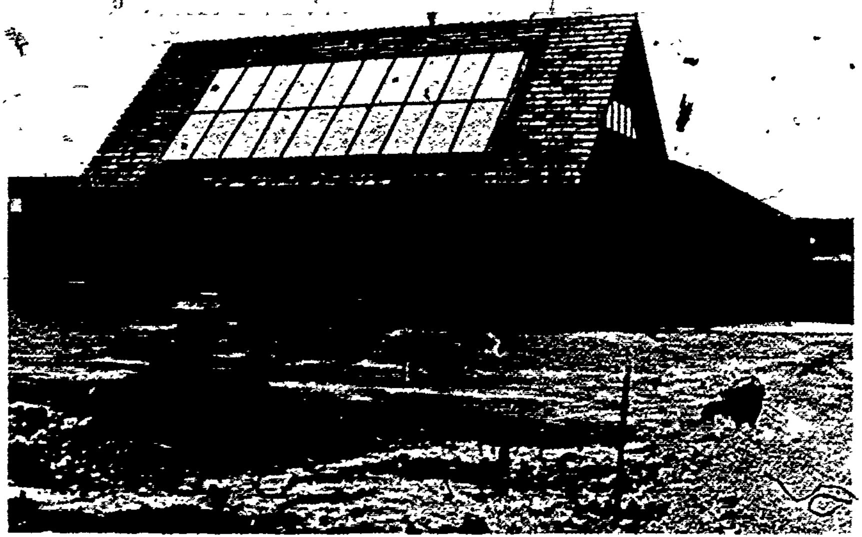

Table 2-6. Design Data for Eco-Era No. 2

Floor Space	1950 Ft^2
Collector Size	390 Ft^2
Collector Type	Flat Plate (Air)
Storage Volume	195 Ft^3
Storage Medium	Pebbles (∿12 tons)
System Cost	∿$11,000

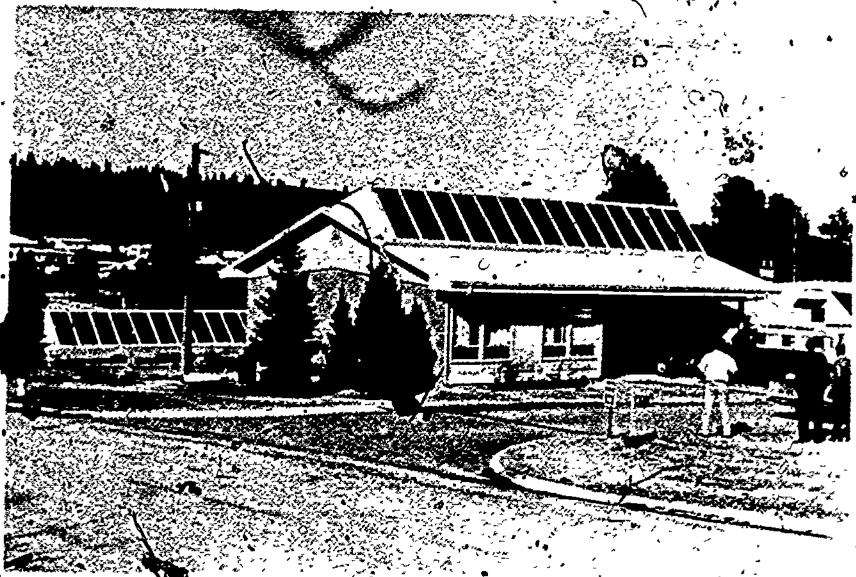

Figure 2-5. USAFA House

Table 2-7. Design Data for USAFA Installation

Floor Space	1665 Ft^2
Collector Size	613 Ft^2
Collector Type	Flat Plate (Liquid)
Storage Volume	2500 Gallons
Storage Medium	Water
System Cost	~$25000

we see the ratio of collector area to floor area is slightly greater than 1 to 3, the ratio of storage volume [illegible]llector size was approximately 4 gallons of water per square foot of collector, and the system cost was approximately $50 per square foot of installed collector. There are many factors that must be taken into consideration for the system cost and this is not to be interpreted as a representative figure for commercial installations. Also, it should be pointed out that based on operational experience gained during the last year, the storage volume was decreased by a factor of approximately 2. This system was completed during the autumn of 1975.

Figure 2-6 shows the Phoenix house in Colorado Springs, Colorado. This house was completed in June 1974. The system is a solar-assisted heat pump arrangement. There are two collector arrays on the house, separated by a flat roof which is covered with white quartz to increase the reflected radiation on the top array of collectors. The transport fluid in the collectors is Dowtherm J, and the storage fluid is water in an underground tank. An air-to-air heat pump was used during the first year of test operation, and was changed to a water-to-air heat pump in 1975. Solar heated water in the large underground tank is used as the primary energy source for the heat pump.

The design data for the Phoenix house are shown in Table 2-8. We observe that the ratio of collector area to floor area is approximately 1 to 3. The storage tank capacity is 8000 gallons, but the tank has been filled to approximately 2500 gallons since the initial investigations were conducted. The ratio of storage volume to collector area is approximately 3 gallons per square foot. The system cost was about $14 per square foot of installed collector.

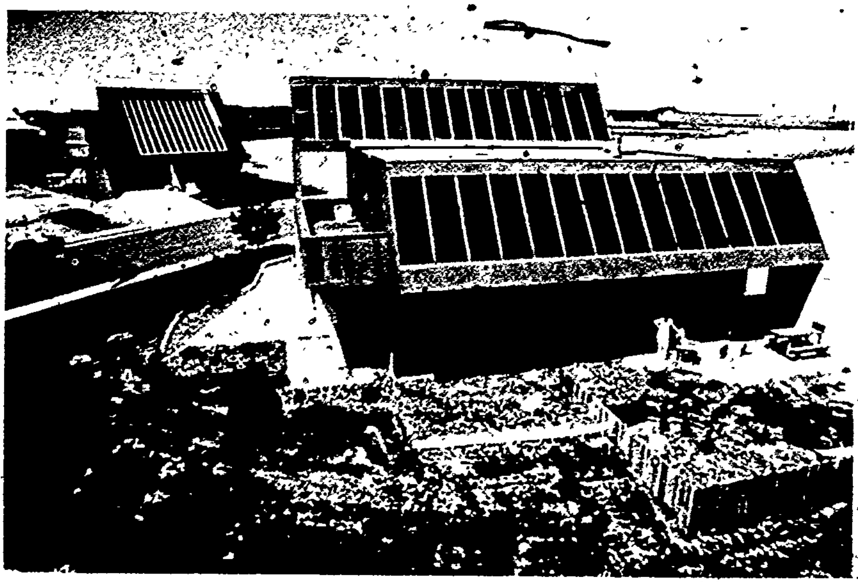

Figure 2-6. Phoenix House in Colorado Springs, Colorado

Table 2-8. Design Data for the Phoenix House

Floor Space	2190 Ft^2
Collector Size	780 Ft^2
Collector Type	Flat Plate (Liquid)
Storage Volume	Variable*
Storage Medium	Water
System Cost	$11122

*. The volume of water in storage has been varied between 2500 and 7000 gallons.

Figure 2-7 shows a liquid-heating solar system installation that was completed in Aspen, Colorado, during the autumn of 1975. The solar system is being used to provide space heating for the pool enclosure (on which the collectors are shown) and heating of the swimming pool. The design data for this installation are shown in Table 2-9. Here we observe a ratio of collector area to floor area of approximately 1 to 4, a ratio of storage volume to collector area of approximately 2 gallons per square foot, and the system cost of $25 per square foot of installed collectors.

Figure 2-8 shows the SECO house that was constructed near Fort Collins, Colorado, during the summer and fall of 1976. The system uses aluminum water-heating collectors, a storage tank and an auxiliary heating unit. The design data for this house are shown in Table 2-10. We observe that the ratio of collector size to floor space is approximately 1 to 5, the ratio of storage volume to collector area is about 1.65 gallons per square foot, and the system cost was nearly $19 per square foot of installed collector.

OPERATION OF AIR SYSTEMS

Figure 2-9 shows a schematic representation of the system in CSU Solar II. This is a representative installation for air systems. The system operation is described below, when solar-heated air is supplied directly to the rooms in the building.

We will trace the flow of air through the system starting from the room air return indicated by the arrow. The air will flow through the duct and be prevented from being exhausted by motorized damper MD4.

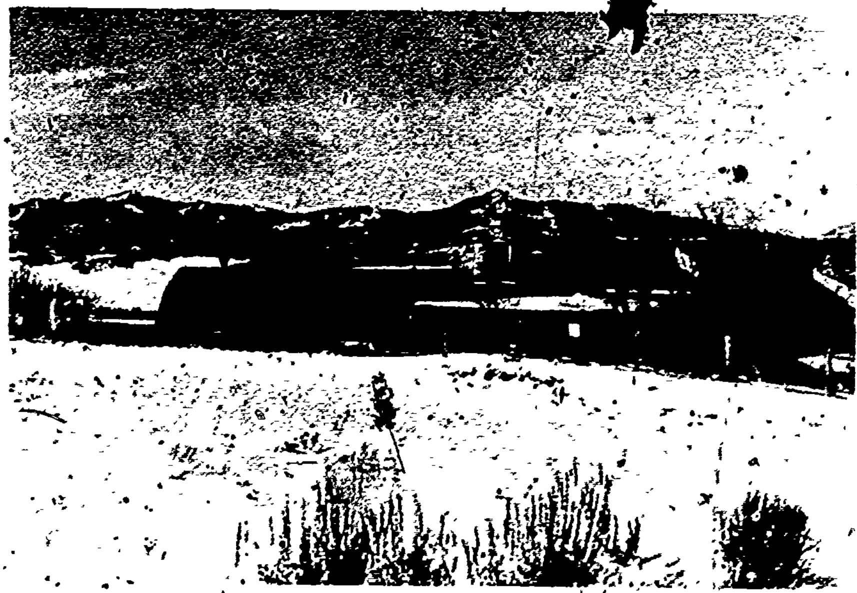

Figure 2-7. Aspen System

Table 2-9. Design Data for Aspen Installation

Floor Space	1840 Ft^2
Collector Size	420 Ft^2
Collector Type	Aluminum (Reynolds)
Storage Volume	~800 Gallons
Storage Medium	Water
System Cost	$10500.

Figure 2-8: SECO House Near Fort Collins, Colorado

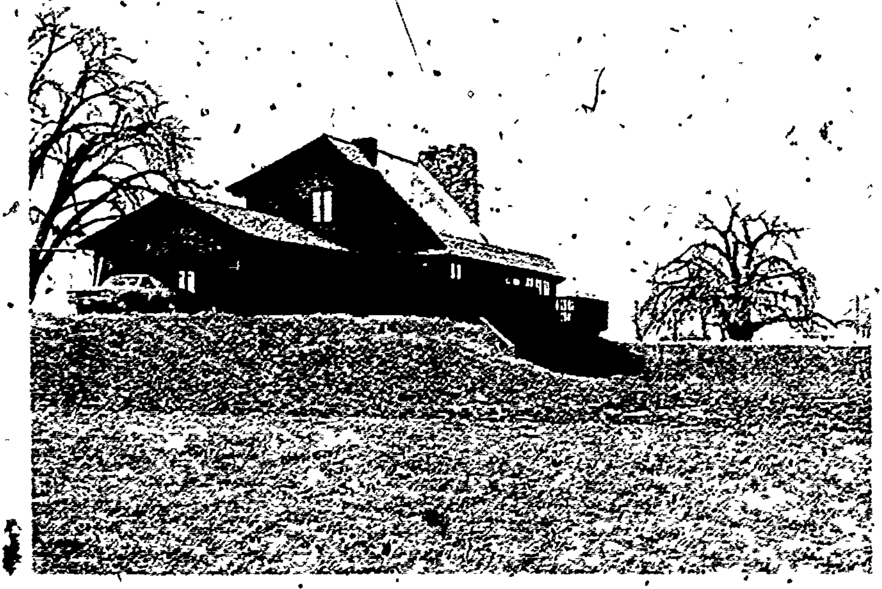

Table 2-10. Design Data for the SECO House

Floor Space	3400 Ft^2
Collector Size	640 Ft^2
Collector Type	Flat Plate (Liquid)
Storage Volume	1056 Gallons
Storage Medium	Water
System Cost	~$12,000

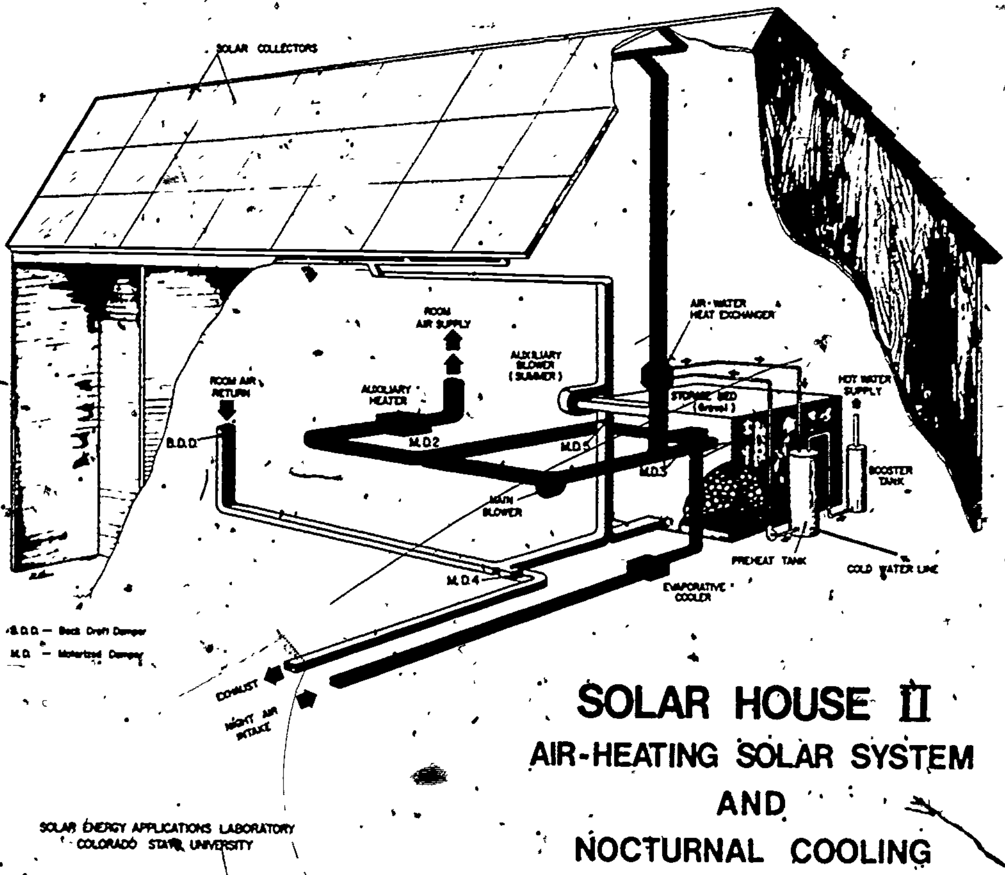
SOLAR COLLECTORS
ROOM AIR SUPPLY
AUXILIARY BLOWER (SUMMER)
AIR-WATER HEAT EXCHANGER
ROOM AIR RETURN
AUXILIARY HEATER
HOT WATER SUPPLY
STORAGE BED (Gravel)
B.D.D.
M.D.2
M.D.1
M.D.3
BOOSTER TANK
MAIN BLOWER
PREHEAT TANK
COLD WATER LINE
M.D.4
EVAPORATIVE COOLER
B.D.D. — Back Draft Damper
M.D. — Motorized Damper
EXHAUST
NIGHT AIR INTAKE
SOLAR HOUSE II
AIR-HEATING SOLAR SYSTEM
AND
NOCTURNAL COOLING
SOLAR ENERGY APPLICATIONS LABORATORY
COLORADO STATE UNIVERSITY

Figure 2-9. Schematic Representation of CSU Solar II System

The flow will then proceed through the other path and be prevented from going through storage by motorized dampers 2 and 3. Motorized damper 5 will be open to allow the air to flow up to the supply manifold shown on the bottom side of the collectors. The air will flow through the collectors and be heated in the process. It will then be returned from the top of the collectors by flowing down through the ducting containing the air-water heat exchanger shown in the figure. Motorized damper number 3 will allow the hot air to flow to the main blower. The air will come out of the main blower and proceed toward motorized damper number 2, which will direct the flow toward the auxiliary heater. If the temperature of the air is sufficiently high to provide the required heating, the auxiliary heater will not be turned on, and the warm air will be supplied to the room as indicated by the arrows labeled room air supply. In the event the air is not heated to a sufficiently high temperature in the collectors, the auxiliary heater will be turned on to boost the temperature of the air being supplied to the room.

In the storage mode, motorized damper 2 will divert the flow of air from the main blower toward the top of the storage bed as indicated in Figure 2-9. The hot air will enter the top of the storage bed, will give up its heat to the pebbles, and will be exhausted from the bottom of the storage bed as indicated. In this case, the back draft damper (BDD) and motorized damper 4 will be closed, and motorized damper 5 will be open. Hence, the air will flow back up to the supply manifold on the collectors, be heated by the collectors and flow into storage to complete the cycle.

In order to heat the enclosure from storage, the room air will enter the system at the point labeled room air return and will flow toward the storage bed. Motorized dampers 4 and 5 will be closed, thereby forcing the cool air into the bottom of the storage bed. It will flow up through the storage bed, be heated in the process and will exit from the top of the storage bed toward motorized damper number 1. Motorized damper 1 will direct the flow toward motorized damper 3, which in turn will direct the flow of air to the main blower. The air will exit the main blower and proceed toward motorized damper 2, which will direct the flow toward the auxiliary heater. From there, the process is the same as that described for heating directly from the collectors.

The service hot water is obtained from the air/water heat exchanger shown in the return air duct from the collectors. The water in the preheat tank is pumped from the preheat tank, through this heat exchanger, and returned to the preheat tank. Hot water that is taken from the booster tank is replaced by water from the preheat tank. During summer operation, when it is not desirable to have heat in the storage bed, the service hot water is obtained in the following manner. Hot air, after passing by the air/water heat exchanger, is diverted toward the auxiliary blower through the small duct shown between the auxiliary blower and the return air duct. The auxiliary blower then supplies air to the supply manifold of the collectors. It is heated in the collectors in order to provide for service hot water.

There are obvious variations on this system that one could develop. For example, it might be desirable to take warm air from the attic of a house to circulate through the collectors and provide for service hot water. This warm air could then be exhausted to the outside and replaced by warm air from the attic. This process would serve to provide for service hot water in the summer and also some cooling of the house by getting hot air out of the attic.

The nocturnal cooling shown in Figure 2-9 operates as follows: During the summer, when cooling may be desired, the rock bed can be cooled at night by bringing cool night air in at the point indicated in Figure 2-9. This cool night air is reduced to wet-bulb temperature by the evaporative cooler and is then sent to the storage bed by passing through motorized damper 3, the main blower, and motorized damper 2. It passes through the storage bed from the top to bottom and is exhausted to the outside by passing through the motorized damper 4. In this case, the back draft damper is closed. If this system operates during most of the night, then there will be cool rocks in the storage bed from which cool air can be obtained the following day. The cool air is obtained by taking warm air in through the room air return, passing it through the storage bed from the bottom to top, then out of the storage bed through motorized damper 3, the main blower, motorized damper 2, and to the room air supply.

A simplified schematic of a typical air system using two blowers is shown in Figure 2-10. The system represented in this figure would require two blowers and two 3-way dampers. For the system as shown, if blower number 1 is turned on, the pebble-bed storage will be heated. By changing the positions of the dampers, the building could be heated directly from the collectors, using blower number 1 and blower number 2, or the building could be heated from storage by using blower number 2 and having the dampers direct the flow through the storage in the opposite direction.

The use of two blowers can be avoided by utilizing an arrangement as shown in Figure 2-11. This system is representative of the system installed in CSU Solar II. The ducting may be more extensive, but a second blower is eliminated.

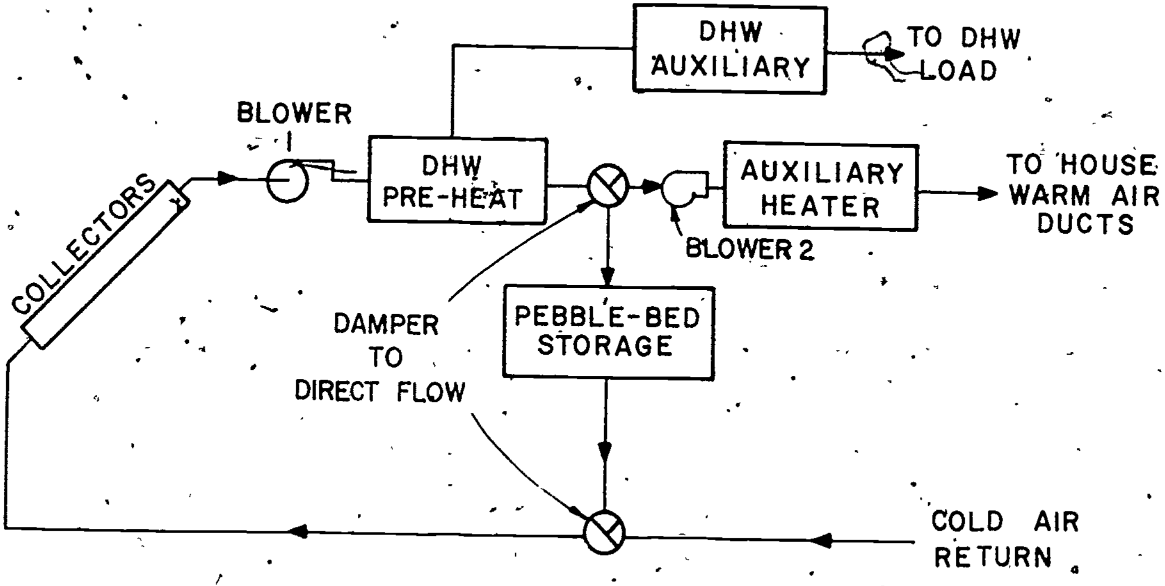
DHW AUXILIARY
TO DHW LOAD
BLOWER 1
DHW PRE-HEAT
AUXILIARY HEATER
TO HOUSE WARM AIR DUCTS
COLLECTORS
BLOWER 2
DAMPER TO DIRECT FLOW
PEBBLE-BED STORAGE
COLD AIR RETURN

Figure 2-10. Simplified Schematic of an Air System

WATER SYSTEMS

Figure 2-12 shows a schematic representation of the solar equipment in CSU Solar I. This represents a typical installation involving water as the transport medium and storage. The heavy lines indicate the flow in the collector loop. The diagram illustrates the situation when heat is being stored from the collectors via the collector heat exchanger. The water is shown coming down from the collectors, going to the collector heat exchangers, out of the collector heat exchangers, and then through the pump labeled P4 and finally back up to the collector.

LEGEND

EVC =EVAPORATIVE COOLER
AUX =AUXILIARY HEATER
HUM =HUMIDIFIER
B1,B2 =BLOWERS
P =RECIRCULATING PUMP
MD =MOTORIZED DAMPER
W/S =MANUAL WINTER/SUMMER DAMPER

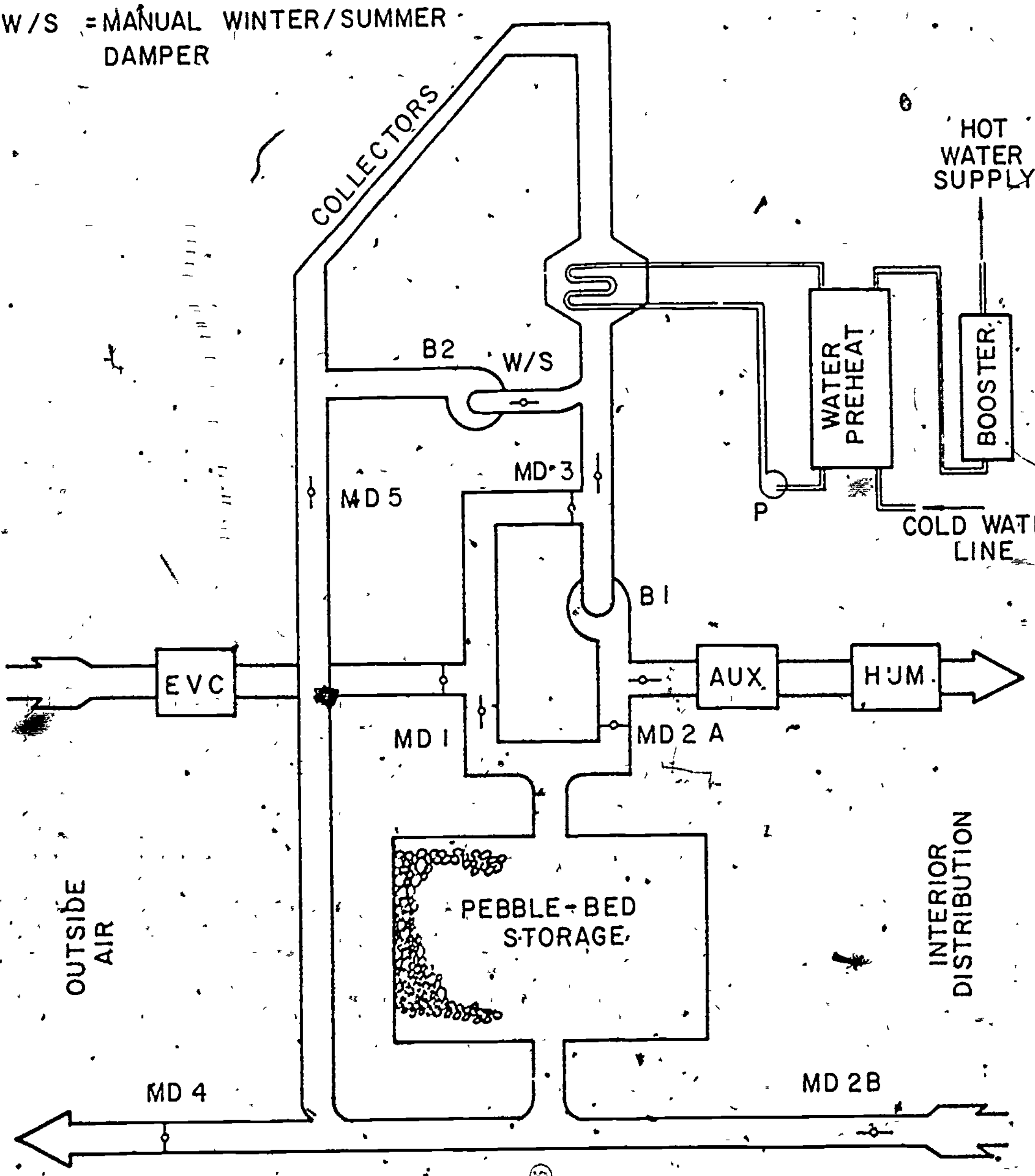
COLLECTORS
HOT WATER SUPPLY
B2
W/S
WATER PREHEAT
BOOSTER
MD 3
MD 5
P
COLD WATER LINE
B1
EVC
AUX
HUM
MD 1
MD 2 A
PEBBLE-BED STORAGE
OUTSIDE AIR
INTERIOR DISTRIBUTION
MD 4
MD 2B

Figure 2-11. Schematic of an Air System

The remaining flow is shown coming out of the bottom of the thermal storage unit, going through pump P3, then through the collector heat exchanger and back to the top of the thermal storage tank. The controls required for accomplishing this task will be discussed later.

Figure 2-13 represents the situation where heating is being supplied from storage or the auxiliary system; again, the heavy lines in this figure represent the flow. The water is shown coming out of the top of the thermal storage unit, going down and over through pump P1, then up through valve V1 to the heating coils. The heating coils are simply a water-to-air heat exchanger, and the output from the heating coils is warm air to be supplied to the house. After passing through the heating coils, the water is directed either back to the thermal storage unit or to the auxiliary hot water boiler by means of valve V2. The auxiliary hot water boiler is utilized if there is not enough heat in storage to satisfy the demand heat load for the house. The system as shown here will obtain heat either from the solar storage or from auxiliary, but not from both simultaneously. The house was constructed so that this latter mode could be utilized, and this will be done sometime in the future in order to compare the different operational strategies.

Figure 2-14 represents the method by which service hot water is obtained from the solar heating. Here, the water is shown coming from the top of the thermal storage tank and to the input side of the hot water heat exchanger, then out of the hot water heat exchanger through pump P2 and back to the thermal storage unit. The other side of the heat exchanger is connected to the 80-gallon hot water preheat tank. In this loop, the flow is out of the preheat tank, then through pump P6, then to the hot water heat exchanger, then out of the hot water heat exchanger and back to the preheat tank. This preheat tank is connected to a standard

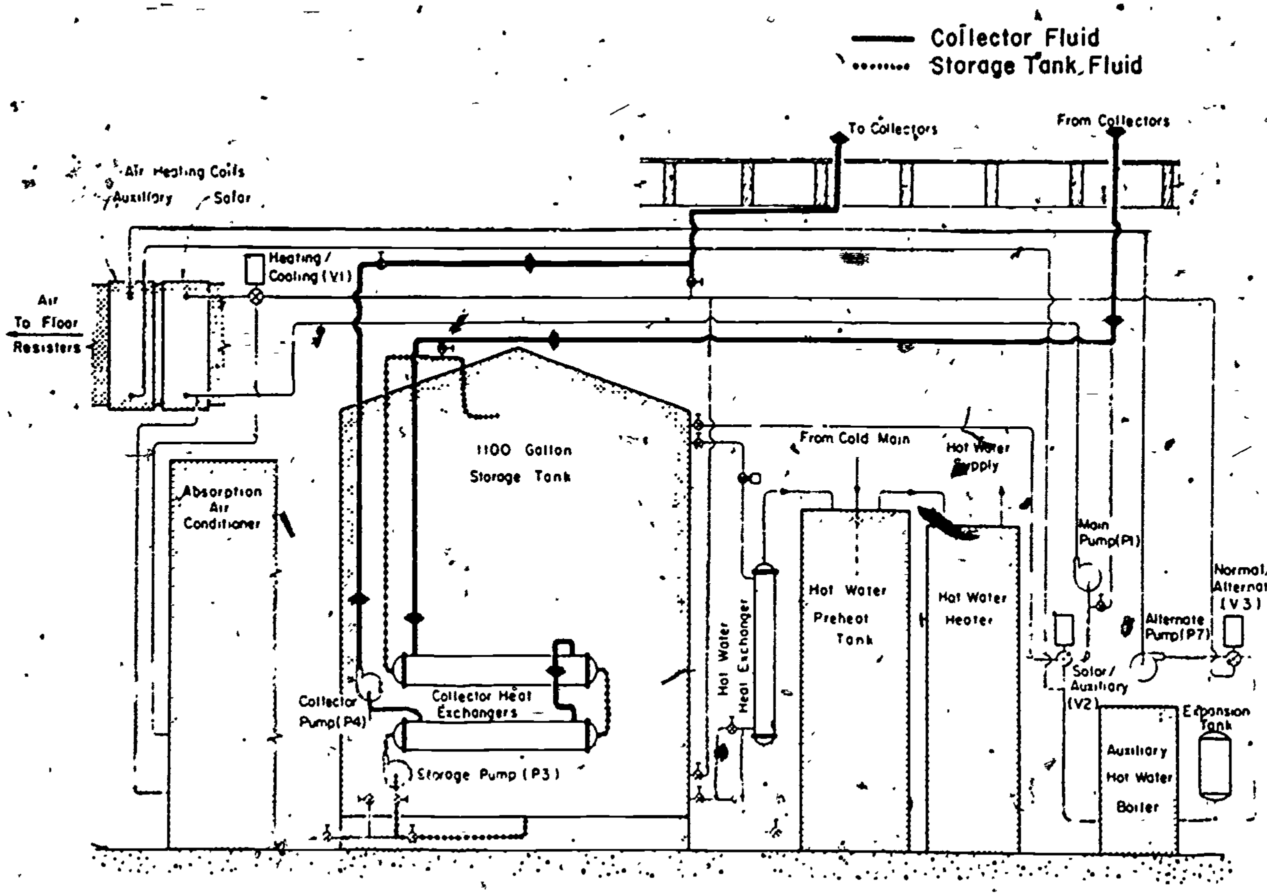
Collector Fluid
Storage Tank Fluid
To Collectors
From Collectors
Air Heating Coils
Auxiliary
Solar
Heating / Cooling (V1)
Air To Floor Resisters
Absorption Air Conditioner
1100 Gallon Storage Tank
From Cold Main
Hot Water Supply
Main Pump (P1)
Normal/Alternate (V3)
Alternate Pump (P7)
Solar/Auxiliary (V2)
Expansion Tank
Auxiliary Hot Water Boiler
Hot Water Preheat Tank
Hot Water Heater
Hot Water Heat Exchanger
Collector Heat Exchangers
Collector Pump (P4)
Storage Pump (P3)

Figure 2-12. Mode 1 - Solar Collection Using Heat Exchanger

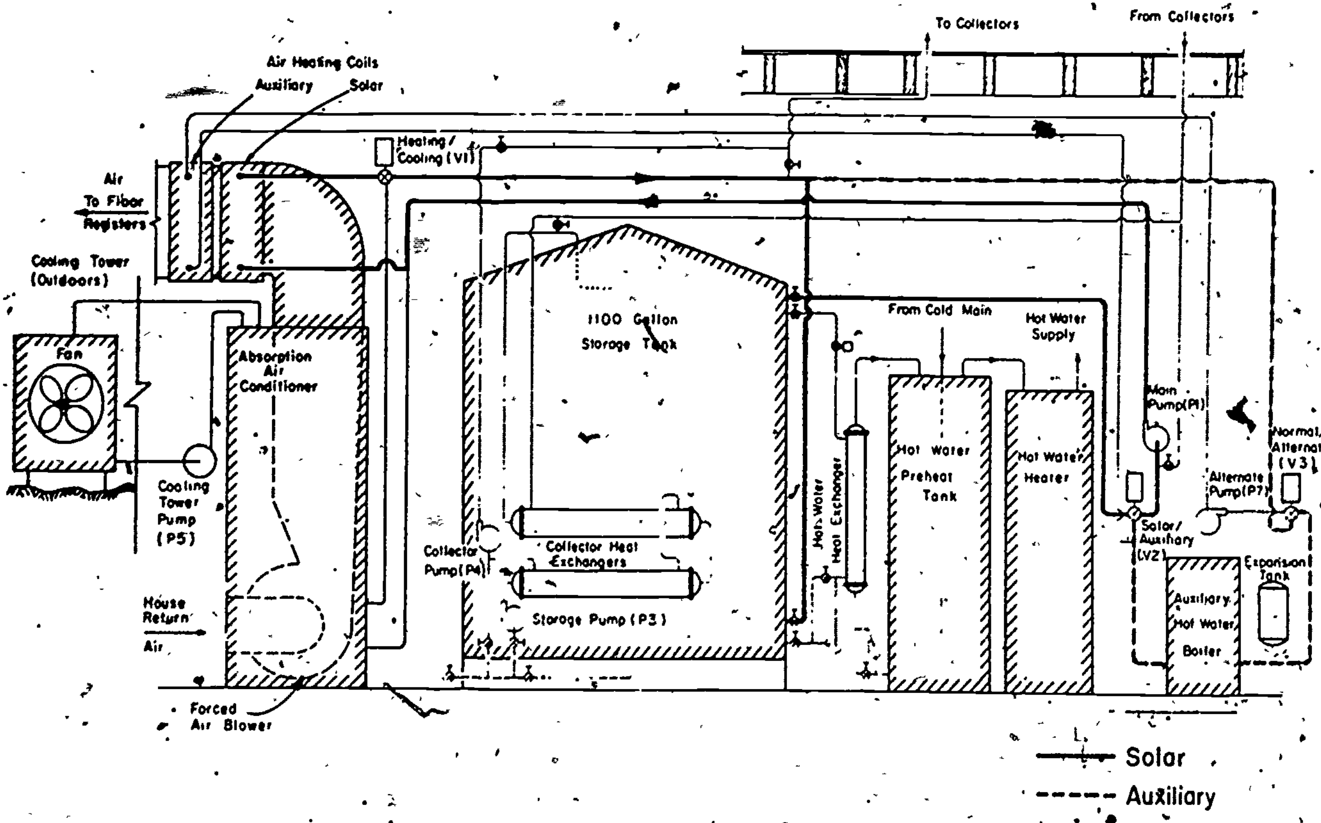
To Collectors
From Collectors
Air Heating Coils
Auxiliary
Solar
Heating / Cooling (V1)
Air To Floor Registers
Cooling Tower (Outdoors)
Fan
Absorption Air Conditioner
Cooling Tower Pump (P5)
House Return Air
Forced Air Blower
1100 Gallon Storage Tank
Collector Pump (P4)
Collector Heat Exchangers
Storage Pump (P3)
Hot Water Heat Exchanger
From Cold Main
Hot Water Supply
Hot Water Preheat Tank
Hot Water Heater
Main Pump (P1)
Normal/Alternate (V3)
Alternate Pump (P7)
Solar/Auxiliary (V2)
Expansion Tank
Auxiliary Hot Water Boiler
Solar
Auxiliary

Figure 2-13. Modes 1 and 2 Space Heating

40-gallon hot water heater. The cold water makeup enters the 80-gallon preheat tank.

Finally, Figure 2-15 shows the case of solar cooling from the storage or the auxiliary boiler unit. This is similar to [illegible] earlier figure that showed the heating situation. In this case, however, valve V1 directs the flow to the lithium bromide absorption refrigeration unit. The flow is then either returned to the thermal storage unit or directed to the auxiliary hot water boiler by valve V2. Again, the auxiliary hot water boiler is utilized when there is not sufficient heat in the storage tank to operate the generator on the air conditioning unit. The output of this system is cool, dehumidified air to be used to cool the house.

Alternative strategies and systems may be used for both heating and cooling. Some of these will be discussed in other modules.

DOMESTIC HOT WATER SYSTEMS

There may be many cases in which it is desirable to have a solar service hot water system and not consider space heating or space cooling. In this section we introduce the concept of solar service hot water systems and present some schematic representations of possible configurations.

The basic components are still the collector array, the storage, a pump or blower, the controller, and possibly a heat exchanger. There are several companies marketing solar service hot water systems at the present time. One may purchase a complete system or, instead, one may elect to purchase the components and assemble the system.

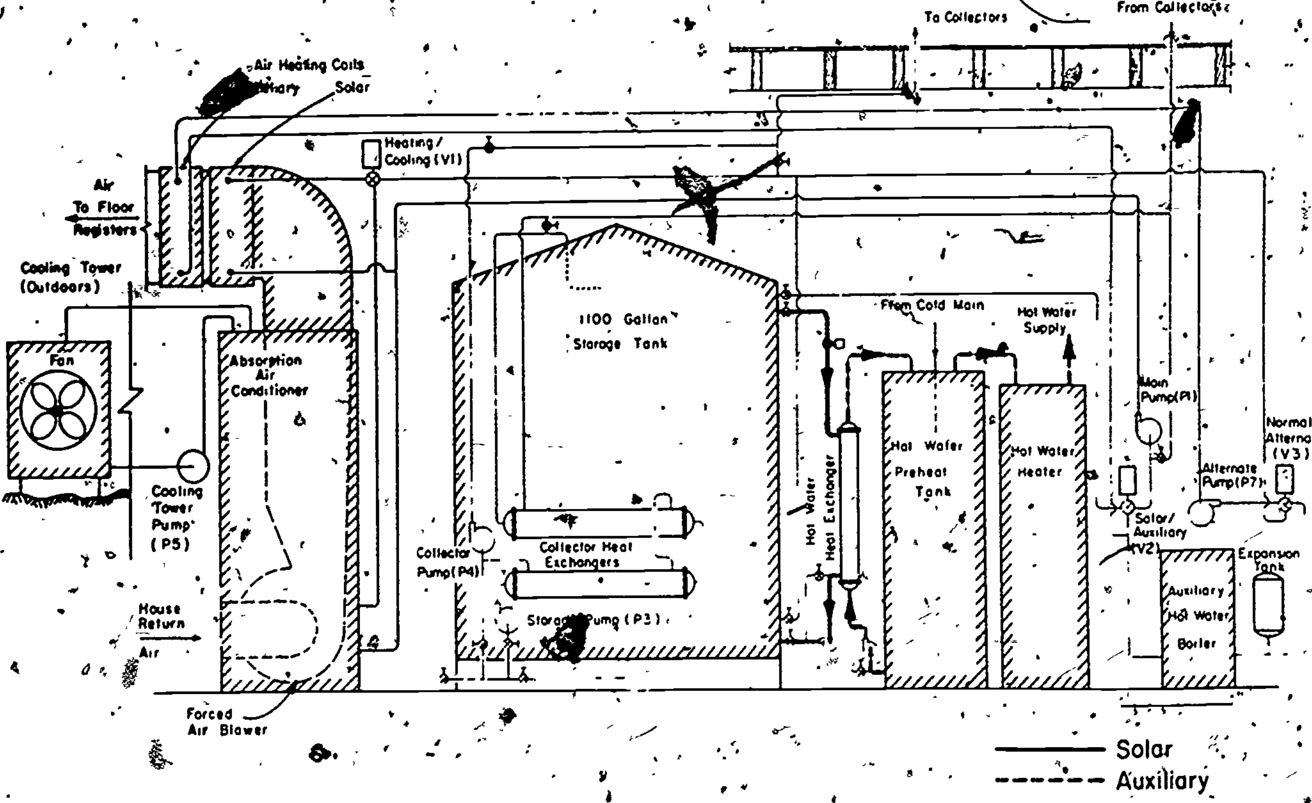
To Collectors
From Collectors
Air Heating Coils
Auxiliary
Solar
Heating / Cooling (V1)
Air To Floor Registers
Cooling Tower (Outdoors)
Fan
Absorption Air Conditioner
1100 Gallon Storage Tank
From Cold Main
Hot Water Supply
Main Pump (P1)
Normal / Alternate (V3)
Alternate Pump (P7)
Hot Water Heat Exchanger
Hot Water Preheat Tank
Hot Water Heater
Solar/ Auxiliary (V2)
Expansion Tank
Auxiliary Hot Water Boiler
Cooling Tower Pump (P5)
Collector Pump (P4)
Collector Heat Exchangers
Storage Pump (P3)
House Return Air
Forced Air Blower
Solar
Auxiliary

Figure 2-14. Solar Heating - Service Hot Water

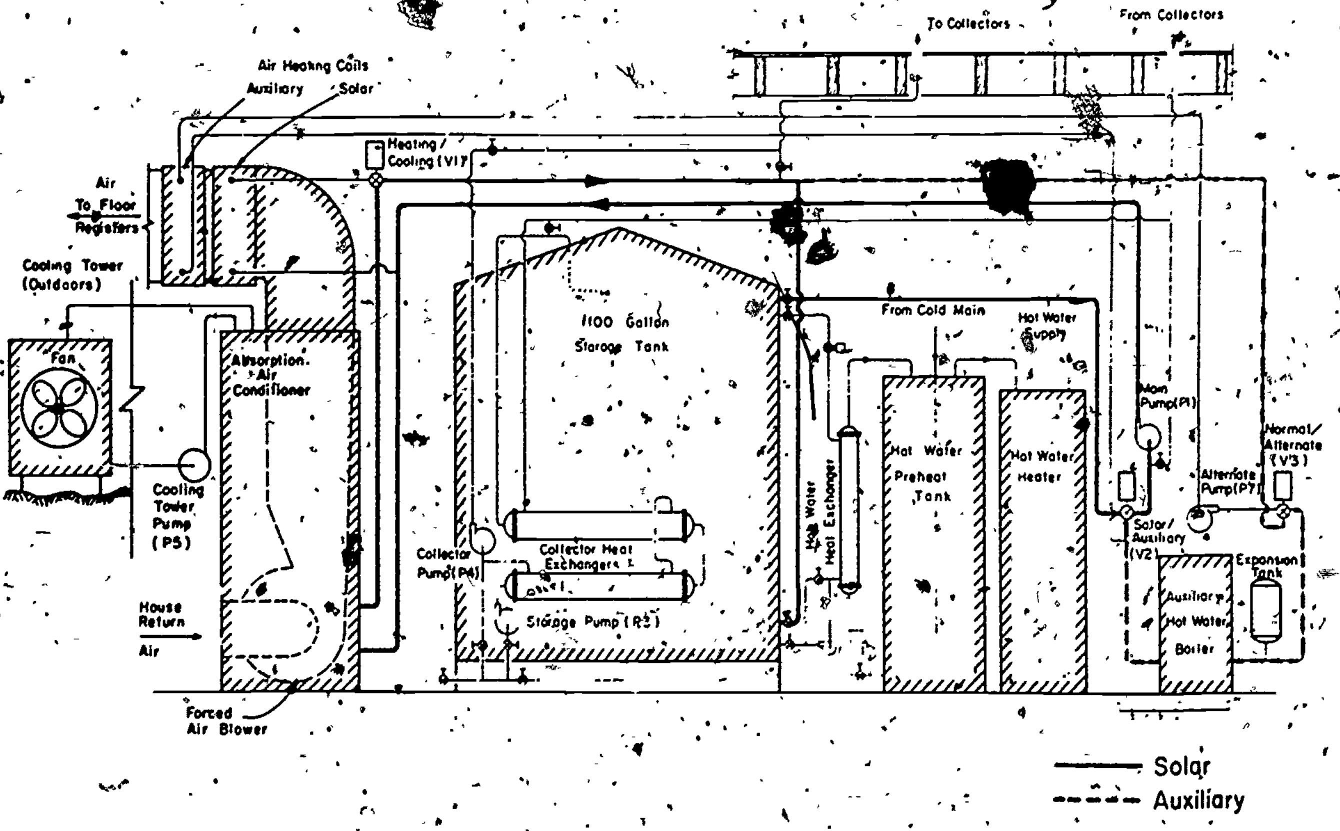
To Collectors
From Collectors
Air Heating Coils
Auxiliary
Solar
Heating / Cooling (V1)
Air To Floor Registers
Cooling Tower (Outdoors)
Fan
Absorption Air Conditioner
1100 Gallon Storage Tank
From Cold Main
Hot Water Supply
Main Pump (P1)
Normal / Alternate (V3)
Alternate Pump (P7)
Hot Water Heat Exchanger
Hot Water Preheat Tank
Hot Water Heater
Solar/ Auxiliary (V2)
Expansion Tank
Auxiliary Hot Water Boiler
Cooling Tower Pump (P5)
Collector Pump (P4)
Collector Heat Exchangers
Storage Pump (P3)
House Return Air
Forced Air Blower
Solar
Auxiliary

Figure 2-15. Modes 1 and 2 Space Cooling

A configuration using a water collector is shown in Figure 2-16. There is no heat exchanger required in this system and there is no corrosion inhibitor or antifreeze solution added to the water. Consequently, this system would not be recommended for cold climates unless the collectors are to be drained when not operating or pulsed whenever the temperature of the water gets near freezing.

Figure 2-17 shows an alternative arrangement for a solar service hot water system. This system would be required in a situation where a non-potable solution has been added to the collector transport medium.

The double-wall heat exchanger effect can be achieved by a system as shown in Figure 2-18. This system would use a fiberglass, glass-lined, or stainless steel storage tank for which no corrosion inhibitor is required. This system is obviously more expensive than those shown previously.

These systems will be discussed in greater detail in a later module.

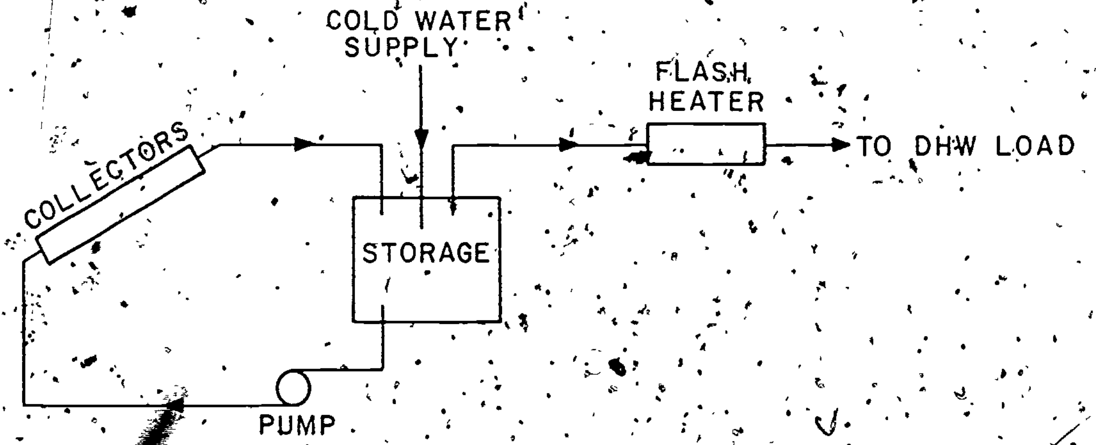
COLD WATER SUPPLY
FLASH HEATER
COLLECTORS
STORAGE
TO DHW LOAD
PUMP

Figure 2-16. Solar Service Hot Water System

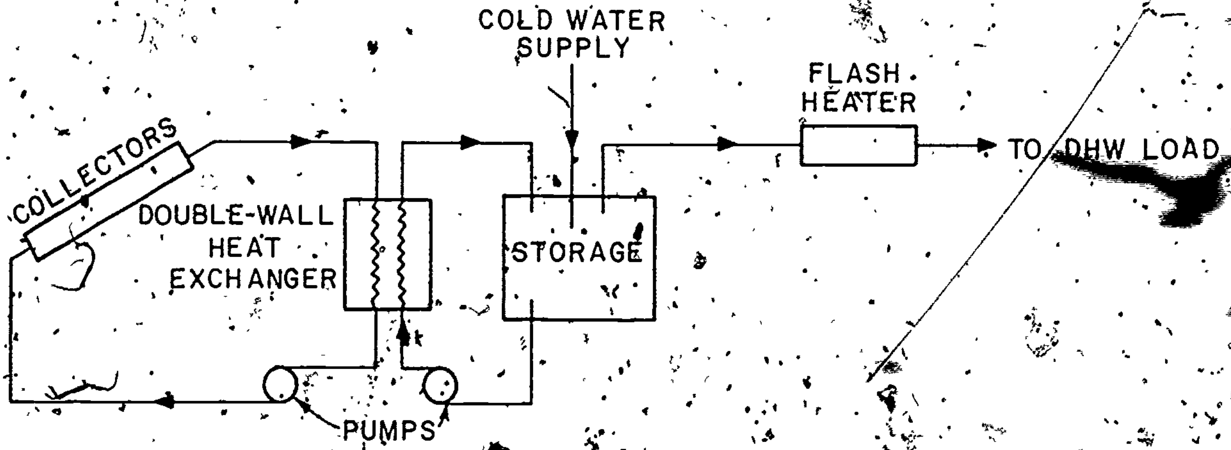
COLD WATER SUPPLY
FLASH HEATER
COLLECTORS
DOUBLE-WALL HEAT EXCHANGER
STORAGE
TO DHW LOAD
PUMPS

Figure 2-17. Solar Service Hot Water System with Heat Exchanger

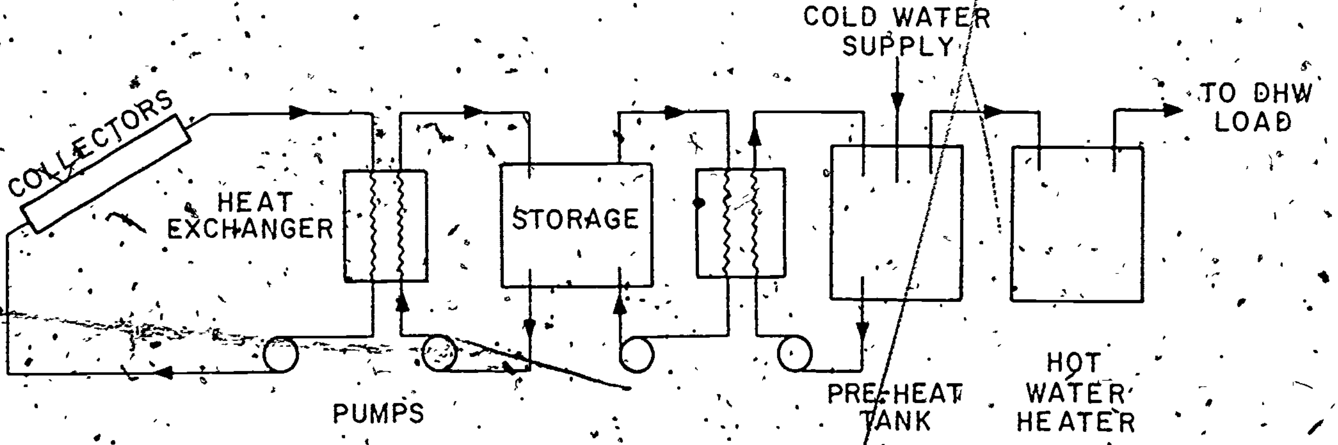
COLLECTORS
HEAT EXCHANGER
STORAGE
PUMPS
COLD WATER SUPPLY
TO DHW LOAD
PRE-HEAT TANK
HOT WATER HEATER

Figure 2-18. Solar Service Hot Water System with Dual Heat Exchangers

TRAINING COURSE IN
THE PRACTICAL ASPECTS OF
DESIGN OF SOLAR HEATING AND COOLING SYSTEMS
FOR
RESIDENTIAL BUILDINGS

MODULE 3

SOLAR RADIATION INFORMATION FOR DESIGN PURPOSES

SOLAR ENERGY APPLICATIONS LABORATORY
COLORADO STATE UNIVERSITY
FORT COLLINS, COLORADO

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GLOSSARY OF TERMS

Beam Radiation	Radiation reaching the earth's surface from the solar disc.
Declination	Position of the sun relative to the equatorial plane at solar noon.
Diffuse Radiation	Radiation reaching the earth's surface from 180 degrees hemisphere, excluding the beam radiation.
Direct Radiation	Radiation reaching the earth's surface from the solar disc.
Solar Radiation	Portion of total radiation which is useful for solar heating and cooling systems.
Solar Weather	Climatic factors which influence solar collection.
Tilted Surface	Tilted with respect to the horizontal.

LIST OF SYMBOLS

$\overline{D}$	Average daily diffuse radiation for a month
$\overline{H}$	Monthly average daily total radiation on a horizontal surface, Btu/(day)(ft^2)
H_o	Extraterrestrial daily radiation on a horizontal surface for the 16th day of the month, Btu/(day)(ft^2)
$\overline{H}_o$	Monthly averaged value of extraterrestrial radiation on a horizontal surface, Btu/(day)(ft^2)
$\overline{H}_T$	Radiation on a tilted surface averaged over a day, Btu/(day)(ft^2)
$\overline{K}_T$	Fraction of solar energy which penetrates through the earth's atmosphere on daily average
n	Number of days from January 1
$\overline{R}$	Fraction of average daily radiation on tilted surface compared with a horizontal surface
R_D	Ratio of the average daily beam radiation on a tilted surface to that on a horizontal surface
S	Collector tilt angle from horizontal, degrees
T	Temperature, °F
t	Time variable
ω_s	Sunset hour angle, degrees from solar noon
δ	Position of the sun relative to the equatorial plane at solar noon, degrees
ϕ	Latitude angle, degrees (north plus)
ρ	Reflectivity of material or ground surface

OBJECTIVE

The objective of this module is for the trainee to be able to determine the radiation on a tilted collector surface by using the charts and figures presented in this module.

THE NATURE OF SOLAR RADIATION

The energy from the sun is derived from thermonuclear reactions in its core. This energy makes its way to the sun's exterior layers from which it is radiated into interplanetary space. The radiation consists of particulate radiation and electromagnetic radiation. The particulate radiation consists of electrons and protons and is commonly referred to as the "solar wind". The electromagnetic radiation is what is commonly referred to as "solar radiation" and it is this radiation that partially penetrates the earth's atmosphere and is utilized in a solar heating or cooling system. This solar radiation varies inversely with distance from the sun. Since the earth's distance from the sun varies by only about three percent during the course of the year, the amount of solar radiation reaching the upper limits of the earth's atmosphere is essentially constant. This is referred to as the "solar constant" and is defined as the solar radiation received on a unit area of surface per unit time perpendicular to the radiation at the earth's mean distance from the sun. Recent measurements of the solar constant have indicated that its value should be 1353 watts per square meter (428 Btu per square foot per hour, 4871 kJ per square meter per hour, or 1.940 calories per square centimeter per minute). (Ref. 1)

The amount of radiation on a surface perpendicular to the solar radiation at the mean distance of the earth from the sun is essentially constant, but the amount of solar radiation that reaches the surface of the earth will vary with respect to latitude and time of year. It is necessary that this "solar weather" be known in order to design a solar system.

RADIATION ON A HORIZONTAL SURFACE

Hottel and Whillier (Ref. 2, 3) first developed a comprehensive approach to a generalized, long-term description of solar weather. Using their "utilizability" method, it is possible to separate the treatment of solar radiation data from the physical and geometrical characteristics of a particular collector. It is not sufficient, for purposes of predicting detailed collector performance, to use only long-term daily or monthly averages of solar radiation. Hourly fluctuations of radiation about these average values must be taken into account. This is accomplished by using hourly radiation data for a particular location over a period of several years to establish a set of radiation distribution and ϕ-curves. The ϕ or "utilizability" function is of central importance and accounts statistically for the effect of fluctuations in solar weather on collector performance. By use of the utilizability function, account is taken of the fact that only a certain average fraction of the incident radiation on a collector can be "utilized", i.e., converted to usable heat at a given collector plate temperature, T_c.

LIU AND JORDAN METHOD

The ϕ-curves of Hottel and Whillier are site-specific; a set of distribution and ϕ-curves must be generated for each location using solar radiation over a three to five year period. Liu and Jordan (Ref. 4, 5) showed that the long-term solar weather at any location can be characterized surprisingly well by just two site-specific parameters:

$\overline{H}$	-	monthly average daily total radiation on a horizontal surface
$\overline{K}_T$	=	$\overline{H}/\overline{H}_0$

where H_0 is the extraterrestrial daily radiation on a horizontal surface, calculated from the equations of solar geometry for the 16th day of each month.

The term, $\overline{K}_T$, introduced by Liu and Jordan, can be considered as a "cloudiness index" which, together with $\overline{H}$, characterizes the solar weather for a particular month. A large value of $\overline{K}_T$ indicates sunny and rather uniform clear weather. A small value indicates cloudy, more fluctuating weather. In most cases, $\overline{K}_T$ is found to lie in the range from about 0.3 up to 0.75.

Values for $\overline{H}$ and $\overline{K}_T$ and average ambient temperature are presented in Table 3-1 for a number of selected locations. These are taken from Reference 4. However, $\overline{K}_T$ was defined in this case as the ratio between the average radiation on a horizontal surface and the extraterrestrial radiation on the sixteenth of each month.

It is necessary that $\overline{H}$ and $\overline{K}_T$ be known in order to calculate collector performance. Collector performance will be discussed in a later module. For now we must content ourselves with learning how to estimate the radiation that will be received by a collector.

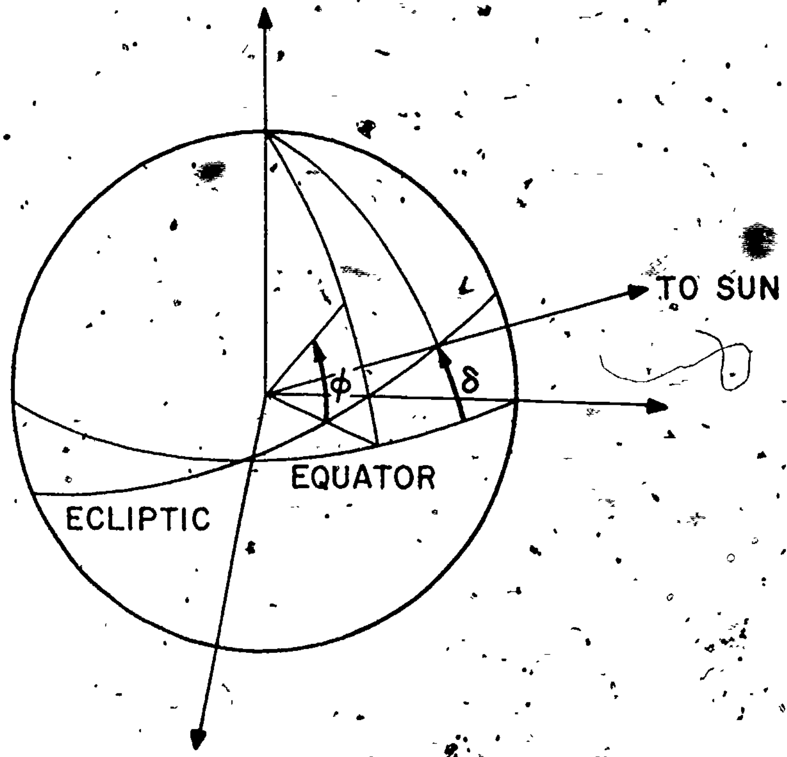
KLEIN, DUFFIE, AND BECKMAN METHOD

The Liu and Jordan approach has been modified by Klein, Duffie, and Beckman. (Ref. 6) In this approach, $\overline{K}_T$ is defined as the ratio between the monthly average daily total radiation on a horizontal surface and the mean daily extraterrestrial radiation, $\overline{H}_o$, where $\overline{H}_o$ may be calculated from the equation:

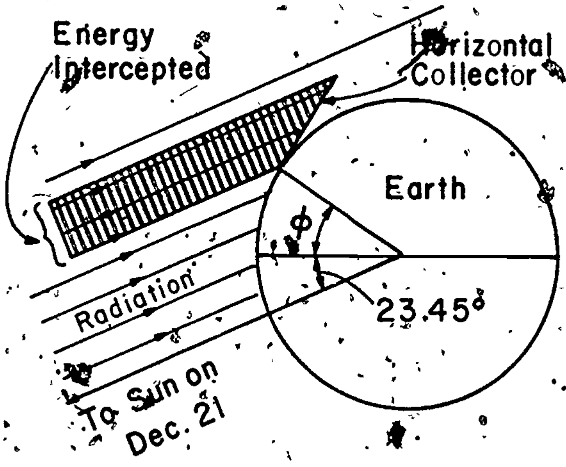
$$\overline{H}_o = \frac{1}{\Delta t} \int_0^{\Delta t} \frac{24}{\pi} I_{sc}\left(1 + 0.033 \cos\left(\frac{360\ n}{365}\right)\right)$$

$$[\cos\phi \cos\delta \sin\omega_s + \omega_s \frac{2\pi}{360} \sin\phi \sin\delta]\, dt \qquad [3\text{-}1]$$

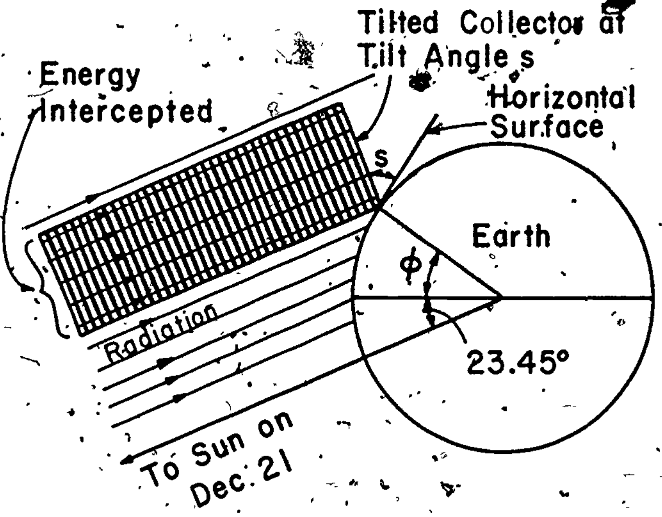
where $\Delta t = 1$ month. I_{sc} is the solar constant, ϕ is the latitude, δ is the solar declination, and ω_s is the sunset hour angle. This is described on the following sketch.



Since the spin axis of the earth is tilted relative to the ecliptic plane, as shown on the sketch, the amount of radiation on a horizontal surface of given area will vary with respect to the time of year. This is illustrated on the sketches below:


Energy Intercepted
Horizontal Collector
Earth
ϕ
Radiation
23.45°
To Sun on Dec. 21

(a) Radiation on a Horizontal Collector


Tilted Collector at Tilt Angle s
Horizontal Surface
Energy Intercepted
s
Earth
ϕ
Radiation
23.45°
To Sun on Dec. 21

(b) Radiation on a Collector Tilted Perpendicular to the Radiation

The declination varies with time of year according to the approximate equation:

$$\delta = 23.45 \sin \left[360 \frac{284 + n}{365}\right] \quad \text{(Degrees)} \qquad [3\text{-}2]$$

where n is the day of the year (n = 1 for January 1). A table of values of δ for the sixteenth of each month is shown on the following page. The sunset hour angle is given by:

$$\cos \omega_s = -\tan \phi \ \tan \delta. \qquad [3\text{-}3]$$

The average daily insolation on a horizontal surface at a given latitude for a given month may be calculated by using the above equations. This has been done for you for various latitudes and the resulting values for H_o for each month are tabulated as a function of latitude in Table 3-2.

Table of Declinations

Date	Day of Year	Declination, δ (Degrees)
Jan. 16	16	-21.10
Feb. 16	47	-12.95
Mar. 16	75	- 2.42
Apr. 16	106	9.78
May 16	136	19.03
June 16	167	23.35
July 16	197	21.35
Aug. 16	228	13.45
Sept. 16	259	1.81
Oct. 16	289	- 9.97
Nov. 16	320	-19.38
Dec. 16	350	-23.37

If $\overline{H}$ is known, then $\overline{K}_T$ may be calculated from $\overline{K}_T = \overline{H}/\overline{H}_o$. Values for $\overline{H}$ may be read from the contour maps showing mean daily solar radiation by month on Figures 3-1 through 3-12.

RADIATION ON A TILTED SURFACE

In order to determine the performance of flat-plate collectors, it is required that the average daily radiation on a tilted surface, $\overline{H}_T$, be known. This may be expressed by $\overline{H}_T = \overline{R}\,\overline{H} = \overline{R}\,\overline{K}_T\,\overline{H}_o$, where $\overline{R}$ is defined as the ratio of the average daily radiation on a tilted surface to that on a horizontal surface for each month. There are several methods for calculating $\overline{R}$, all of which give slightly different results but do not strongly affect the calculations of long-term performance of flat-plate collectors.

One method, proposed by Liu and Jordan, is to express $\overline{R}$ by:

$$\overline{R} = (1 - \frac{\overline{D}}{\overline{H}}) R_D + \frac{\overline{D}}{\overline{H}} \left(\frac{1+\cos s}{2}\right) + \rho \left(\frac{1-\cos s}{2}\right) \qquad [3\text{-}4]$$

where $\overline{D}/\overline{H}$ is obtained from: (Ref. 6)

$$\overline{D}/\overline{H} = 1.3903 - 4.0273\, \overline{K}_T + 5.541\, \overline{K}_T^2 - 3.108\, \overline{K}_T^3 \qquad [3\text{-}5]$$

Also,

$$R_D = \frac{\cos(\phi - s)\cos\delta \sin\omega_s' + \omega_s' \sin(\phi - s)\sin\delta}{\cos\phi \cos\delta \sin\omega_s + \omega_s \sin\phi \sin\delta} \qquad [3\text{-}6]$$

where

$$\omega_s' \triangleq [\arccos(-\tan\phi \tan\delta), \arccos(-\tan(\phi - s)\tan\delta)] \qquad [3\text{-}7]$$

In the above equations,

s is the collector tilt angle from the horizontal

ρ is ground reflectance

$\overline{D}$ is average daily diffuse radiation for each month

R_D is ratio of the average daily beam radiation on the tilted surface to that on a horizontal surface for each month.

It is not necessary that you perform these calculations. Average values of $\overline{R}$ for each month are presented in Tables 3-3 through 3-6. Cross-plots of $\overline{R}$ versus $\overline{K}_T$ obtained from these tables are shown in Figures 3-13 through 3-46. Finally, values for the average radiation on a tilted surface, $\overline{H}_T$, for various values of $\overline{K}_T$ and for collectors at various tilt angles, are shown on Tables 3-7 through 3-22.

EXAMPLES

Examples illustrating the use of these tables and curves are given below:

1. Determine the average daily extraterrestrial radiation for January on a horizontal surface at Columbus, Ohio.

   Solution: From Table 3-1 we see that $\overline{H}$ = 486.3 Btu/ft^2-day; $\overline{K}_T$ = 0.356. Therefore, $\overline{H}_o = \overline{H}/\overline{K}_T$ = 1366 Btu/ft^2-day.

2. Determine the average daily extraterrestrial radiation for January on a horizontal surface at 40°N latitude.

   Solution: From Table 3-2, $\overline{H}_o$ = 1326 Btu/ft^2-day. This compares reasonably well with the earlier result.

3. Determine $\overline{K}_T$ for January for Columbus, Ohio. Use Table 3-2 and the contour maps.

   Solution: Columbus is at 40°N latitude. Hence, from Table 3-2 we see that the average daily extraterrestrial radiation is $\overline{H}_o$ = 1326 Btu/ft^2. Then from Figure 3-1 we see that the average daily radiation on a horizontal surface for January at Columbus is $\overline{H}$ = 128 Langleys = 128 cal/cm^2. If we multiply this by 3.69 we obtain $\overline{H}$ = 472 Btu/ft^2. Therefore, $\overline{K}_T = \overline{H}/\overline{H}_o$ = (472 Btu/ft^2)/(1326 Btu/ft^2) = 0.356.

4. Determine the average daily solar radiation on a surface at tilt angles of 25°, 40°, 55°, and 90° for January at Columbus, Ohio.

a. Tilt = 25°: From Figure 3-17 for latitude of 40°N and tilt = latitude - 15°, we read for $\overline{K}_T = 0.356$, $\overline{R} = 1.395$. Therefore $\overline{H}_T = \overline{R}\,\overline{H} = (1.395)(472) = 658$ Btu/ft^2.

b. Tilt = 40°: From Figure 3-26, $\overline{R} = 1.55$. Therefore $\overline{H}_T = (1.55)(472) = 732$ Btu/ft^2.

c. Tilt = 55°: From Figure 3-35, $\overline{R} = 1.61$. Therefore $\overline{H}_T = (1.61)(472) = 760$ Btu/ft^2.

d. Tilt = 90°: From Figure 3-44, $\overline{R} = 1.43$. Therefore $\overline{H}_T = (1.43)(472) = 675$ Btu/ft^2.

5. Repeat example 4 only use Tables 3-7 through 3-14. Since $\overline{K}_T = 0.356$, we will have to interpolate between the tables for $\overline{K}_T = 0.3$ and for $\overline{K}_T = 0.4$.

a. Tilt = 25°: From Table 3-7, $\overline{H}_T = 536$ Btu/ft^2, $\overline{K}_T = 0.3$. From Table 3-11, $\overline{H}_T = 763$ Btu/ft^2, $\overline{K}_T = 0.4$. The interpolation equation is:

$$\frac{\overline{H}_T - 536}{763 - 536} = \frac{0.356 - 0.3}{0.4 - 0.3}$$

Therefore, $\overline{H}_T = 536 + (227)(.56) = 663$ Btu/ft^2. This agrees to within one percent of the previous result.

b. Tilt = 40°: From Table 3-8, $\overline{H}_T = 584$ Btu/ft^2, $\overline{K}_T = 0.3$. From Table 3-12, $\overline{H}_T = 853$ Btu/ft^2, $\overline{K}_T = 0.4$. Therefore $\overline{H}_T = 584 + (269)(.56) = 735$ Btu/ft^2.

c. Tilt = 55°: From Table 3-9, $\overline{H}_T = 600$ Btu/ft^2, $\overline{K}_T = 0.3$. From Table 3-13, $\overline{H}_T = 890$ Btu/ft^2, $\overline{K}_T = 0.4$. Therefore $\overline{H}_T = 600 + (290)(.56) = 762$ Btu/ft^2.

d. Tilt = 90°: From Table 3-10, $\overline{H}_T = 524$ Btu/ft^2, $\overline{K}_T = 0.3$. From Table 3-14, $\overline{H}_T = 800$ Btu/ft^2, $\overline{K}_T = 0.4$. Therefore $\overline{H}_T = 524 + (276)(.56) = 679$ Btu/ft^2.

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2. Hottel, H.C. and Whillier, A., Evaluation of Flat-Plate Collector Performance. Transactions of the Conference on the Use of Solar Energy: The Scientific Basis, II, Part 1, Section A, 74-104 (1955).

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4. Liu, B.Y.H. and Jordan, R.C., "A Rational Procedure for Predicting a Long-Term Average Performance of Flat-Plate Collectors", Solar Energy, Vol. 17, No. 2, 1963.

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6. Klein, S.A., Beckman, W.A., and Duffie, J.A., "A Design Procedure for Solar Heating Systems". Presented at the International Solar Energy Society Meeting, Los Angeles, California, July/August 1975.

Table 3-1.*

Radiation and Other Data for 80 Locations in the United States and Canada

(H = Monthly average daily total radiation on a horizontal surface, Btu/day-ft², K_t = the fraction of the extra terrestrial radiation transmitted through the atmosphere; t_0 = ambient temperature, deg F.)

		Jan	Feb	Mar	Apr	May	Jun	July	Aug	Sep	Oct	Nov	Dec
Albuquerque, N. M.	H	1150 9	1453 9	1925 4	2343.5	2560.9	2757 5	2561.2	2387 8	2120.3	1639.8	1274.2	1051.6
Lat. 35°03′ N.	K_t	0.704	0 691	0.719	0.722	0.713	0 737	0.695	0.708	0 728	0.711	0 684	0.704
El. 5314 ft	t_0	37.3	43.3	50.1	59.6	69.4	79.1	82.8	80.6	73 6	62.1	47.8	39.4
Annette Is., Alaska	H	236.2	428.4	883.4	1357 2	1634.7	1638.7	1632.1	1269.4	962	454.6	220.3	152
Lat. 55°02′ N.	K_t	0.427	0.415	0.492	0.507	0.484	0.441	0.454	0.427	0 449	0.347	0.304	0.361
El 110 ft	t_0	35.8	37.5	39.7	44.4	51.0	56.2	58.6	59.8	54.8	48.2	41.9	37.4
Apalachicola, Florida	H	1107	1378 2	1654 2	2040.9	2268.6	2195.[illegible]	1978 6	1912.9	1703.3	1544.6	1243.2	982.3
Lat. 29°45′ N.	K_t	0.577	0 584	0 576	0.612	0.630	0.594	0.542	0.558	0.559	0 608	0.574	0.543
El 35 ft	t_0	57.3	59.0	62.9	69.5	76.4	81.8	83 1	83.1	80.6	73.2	63.7	58.5
Astoria, Oregon	H	338.4	607	1008 5	1401.5	1838.7	1753.5	2007.7	1721	1322 5	780.4	413.6	295.2
Lat. 46°12′ N.	K_t	0 330	0.397	0 454	0.471	0.524	0 466	0 551	0.538	0.526	0.435	0.336	0 332
El. 8 ft	t_0	41.3	44.7	46.9	51.3	55.0	59.3	62.6	63.6	62 2	55.7	48.5	43.9
Atlanta, Georgia	H	848	1080 1	1426 9	1807	2018.1	2102 6	2002 9	1898 1	1519.2	1290.8	997.8	751.6
Lat 33°39′ N.	K_t	0 493	0 496	0 522	0.551	0 561	0 564	0 545	0 559	0 515	0 543	0 510	0 474
El. 976 ft	t_0	47.2	49 6	55.9	65.0	78.2	80.9	82.4	81.6	77.4	66.5	54.8	47.7
Barrow, Alaska	H	13.3	143 2	713 3	1491.5	1883	2055.3	1602.2	953 5	428.4	152.4	22.9	—
Lat. 71°20′ N.	K_t	—	0 776	0 773	0 726	0 553	0.533	0 448	0 377	0.315	0 35	—	—
El. 22 ft	t_0	−13 2	−15.9	−12.7	2.1	20 5	35.4	41.6	40 0	31.7	18.6	2.6	−8.6
Bismarck, N D.	H	587 4	934.3	1328 4	1668.2	2056.1	2173 8	2305 5	1929.4	1441 3	1018.1	600.4	464.2
Lat 46°47′ N.	K_t	0 594	0 628	0 605	0 565	0.588	0.579	0.634	0.606	0 581	0 584	0 510	0.547
El 1660 ft	t_0	12.4	15 9	29.7	46.6	58.6	67.9	76.1	73.5	61 6	49.6	31.4	18.4
Blue Hill, Mass.	H	555.3	797	1143.9	1438	1776.4	1943.9	1881.5	1622.1	1314	941	592.2	482.3
Lat. 42°13′ N.	K_t	0 445	0 458	0 477	0 464	0 501	0.516	0 513	0.495	0.492	0.472	0.406	0.436
El 629 ft	t_0	28.3	28.3	36.9	46.9	58.5	67 2	72.3	70.6	64.2	54.1	43.3	31.5
Boise, Idaho	H	518.8	884.9	1280.4	1814.4	2189 3	2376.7	2500 3	2149.4	1717.7	1128.4	678.6	456 8
Lat 43°34′ N.	K_t	0.446	0 533	0 548	0 594	0 619	0.631	0 684	0.660	0.656	0 588	0.494	0.442
El 2844 ft	t_0	29.5	36.5	45.0	53 5	62.1	69.3	79 6	77.2	66.7	56.3	42.3	33.1
Boston, Mass.	H	505.5	738	1067.1	1355	1769	1864	1860.5	1570.1	1267.5	896.7	635.8	442.8
Lat. 42°22′ N	K_t	0 410	0 426	0 445	0.438	0 499	0.495	0.507	0.480	0.477	0.453	0.372	0.400
El. 29 ft.	t_0	31 4	31.4	39.9	49.5	60.4	69.8	74.5	73.8	66.8	57 4	46.6	34.9
Brownsville, Texas	H	1105 9	1262 7	1505 9	1714	2092 2	2288 5	2345	2124	1774.9	1536.5	1104.8	982.3
Lat 25°55′ N.	K_t	0 517	0 500	0.505	0 509	0 584	0 627	0.650	0 617	0 566	0.570	0.468	0.488
El 20 ft	t_0	63.3	66.7	70.7	76.2	81.4	85.1	86.5	86.9	84.1	78.9	70.7	65.2
Caribou, Maine	H	497	861.6	1360 1	1495 9	1779.7	1779.7	1898.1	1675.6	1254 6	793	415.5	398 9
Lat 46°52′ N.	K_t	0 504	0 579	0.619	0 507	0.509	0 473	0.522	0.527	0.506	0 455	0 352	0.470
El 628 ft	t_0	11.5	12.8	24.4	37.3	51.8	61.6	67.2	65.0	56.2	44.7	31.3	16.8
Charleston, S C.	H	946.1	1152 8	1352 4	1918.8	2063.4	2113.3	1649.4	1933.6	1557.2	1332.1	1073 8	952
Lat. 32°54′ N.	K_t	0 541	0.521	0.491	0 584	0.574	0.567	0.454	0.569	0.525	0.554	0.539	0.586
El 46 ft	t_0	53.6	55.2	60.6	67.8	74.8	80.9	82.9	82.3	79.1	69.8	59.8	54.0
Cleveland, Ohio	H	466.8	681.9	1207	1443.9	1928.4	2102.6	2094.4	1840.6	1410.3	997	526.6	427.3
Lat 41°24′ N.	K_t	0.361	0 383	0.497	0.464	0.543	0.559	0.571	0.559	0.524	0.491	0.351	0.371
El. 805 ft.	t_0	30.8	30 9	39.4	50.2	62.4	72.7	77.0	75.1	68.5	57.4	44.0	32.8
Columbia, Mo.	H	651.3	941 3	1315.8	1631.3	1999.6	2129.1	2148.7	1953.1	1689.6	1202.6	839.5	590.4
Lat. 38°58′ N.	K_t	0.458	0.492	0.520	0.514	0.559	0.566	0.585	0.588	0.606	0.562	0.510	0.457
El. 785 ft	t_0	32.5	36.5	45.9	57.7	66.7	75.9	81.1	79.4	71.9	61.4	46.1	35.8
Columbus, Ohio	H	486.3	746.5	1112.5	1480.8	1839.1	(2111)	2041.3	1572.7	1189.3	919.5	479	430.2
Lat. 40°00′ N.	K_t	0.356	0.401	0.447	0.470	0.515	(0.561)	0.555	0.475	0.433	0.441	0.302	0.351
El. 833 ft	t_0	32.1	33.7	42.7	53.5	64.4	74.2	78	75.9	70.1	58	44.5	34.0
Davis, Calif.	H	599.2	945	1504	1959	2368.6	2619.2	2565.6	2287.8	1856.8	1288.5	795.6	550.5
Lat. 38°33′ N.	K_t	0.416	0.490	0.591	0.617	0.662	0.697	0.697	0.687	0.664	0.598	0.477	0.421
El. 51 ft	t_0	47.6	52.1	56.8	63.1	69.6	75.7	81	79.4	76.7	67.8	57	48.7
Dodge City, Kan.	H	953.1	1186.3	1565.7	1975.6	2126.5	2459.8	2400.7	2210.7	1841.7	1421	1065.3	873.8
Lat. 37°46′ N.	K_t	0.639	0.598	0 606	0.618	0.594	0.655	0.652	0.663	0.654	0.650	0.625	0.652
El. 2592 ft	t_0	33.8	38.7	46.5	57.7	66.7	77.2	83.8	82.4	73.7	61.7	46.6	36.8
East Lansing, Michigan	H	425.8	739.1	1086	1249.8	1732.8	1914	1884.5	1627.7	1303.3	891.5	473.1	379.7
Lat. 42°44′ N.	K_t	0.35	0.431	0.456	0.406	0.489	0.508	0.614	0.498	0.493	0.456	0.333	0.349
El. 856 ft	t_0	26.0	26.4	35.7	48.4	59.8	70.3	74.5	72.4	65.0	53.5	40.0	29.0

* Liu, B.Y.H. and Jordan, R.C., "A Rational Procedure for Predicting The Long-Term Average Performance of Flat-Plate Solar-Energy Collectors," Solar Energy, Vol. 7, No. 2, pp. 71-74, 1963.

Table 3-1 (continued)

		Jan	Feb	Mar	Apr	May	Jun	July	Aug	Sep	Oct	Nov	Dec
East Wareham, Mass.	$\bar{H}$	504.4	762.4	1132.1	1392.6	1704 8	1958.3	1873.8	1607.4	1363.8	996.7	636 2	521
Lat. 41°46′ N.	$\bar{K}_t$	0.398	0.431	0.469	0.449	0.480	0.520	0.511	0.489	0.508	0.496	0.431	0 461
El. 18 ft	t_0	32.2	31.6	39.0	48.3	58.9	67.5	74.1	72.8	65.9	56	46	34.8
Edmonton, Alberta	$\bar{H}$	331.7	652.4	1165.3	1541.7	1900.4	1914.4	1964.9	1528.	1113.3	704.4	413.6	245
Lat. 53°35′ N.	$\bar{K}_t$	0.529	0.585	0.624	0.564	0.558	0.514	0.549	0.506	0.506	0.504	0.510	0.492
El. 2219 ft	t_0	10.4	14	26.3	42.9	55.4	61 3	66.6	63.2	54.2	44.1	26.7	14.0
El Paso, Texas	$\bar{H}$	1247.6	1612.9	2048.7	2447.2	2673	2731	2391.1	2350.5	2077.5	1704.8	1324 7	1051.6
Lat. 31°48′ N.	$\bar{K}_t$	0.686	0.714	0.730	0.741	0.743	0 733	0.652	0.669	0.693	0.695	0.647	0.626
El. 3916 ft	t_0	47.1	53.1	58 7	67.3	75.7	84.2	84.9	83.4	78.5	69.0	56.0	48.5
Ely, Nevada	$\bar{H}$	871.6	1255	1749 8	2103.3	2322 1	2649	2417	2307.7	1935	1473	1078.6	814 8
Lat. 39°17′ N.	$\bar{K}_t$	0 618	0 660	0.692	0 664	0.649	0 704	0.656	0.695	0 696	0.691	0 658	0.64
El. 6262 ft	t_0	27.3	32.1	39.5	48.3	57.0	65 4	74.5	72.3	63.7	52.1	39.9	31 1
Fairbanks, Alaska	$\bar{H}$	66	283.4	860.5	1481.2	1806.2	1970.8	1702 9	1247 6	699 6	323.6	104 1	20 3
Lat 64°49′ N.	$\bar{K}_t$	0.639	0.556	0.674	0 647	0.546	0.529	0 485	0 463	0.419	0.416	0.47	0.458
El. 436 ft	t_0	−7.0	0.3	13 0	32 2	50 5	62.4	63 8	58.3	47.1	29.6	5.5	−6 6
Fort Worth, Texas	$\bar{H}$	936 2	1198 5	1597.8	1829.1	2105 1	2437 6	2293 3	2216 6	1880 8	1476	1147 6	913 6
Lat. 32°50′ N.	$\bar{K}_t$	0 530	0 541	0 577	0 556	0.585	0.654	0.624	0.653	0 634	0.612	0.576	0 563
El 544 ft.	t_0	48.1	52.3	59.8	68.8	75.9	84.0	87 7	88.6	81.3	71.5	58 8	50.8
Fresno. Calif	$\bar{H}$	712 9	1116 6	1652.8	2049.4	2409.2	2641.7	2512 2	2300 7	1897 8	1415.5	906.6	616 6
Lat 36°46′ N.	$\bar{K}_t$	0 462	0 551	0 632	0 638	0.672	0.703	0 682	0.686	0 665	0.635	0.512	0.44
El 331 ft.	t_0	47 3	53.9	59 1	65 6	73 5	80 7	87 5	84.9	78.6	68.7	57.3	48 9
Gainesville, Fla.	$\bar{H}$	1036.9	1324 7	1635	1956.4	1934 7	1960 9	1895.6	1873.8	1615 1	1312.2	1169.7	919 5
Lat. 29°39′ N.	$\bar{K}_t$	0 535	0.56	0 568	0.587	0.538	0 531	0.519	0.547	0 529	0.515	0.537	0.508
El. 165 ft	t_0	62.1	63.1	67.5	72.8	79.4	83.4	83.8	84.1	82	75.7	67.2	62 4
Glasgow, Mont.	$\bar{H}$	572 7	965.7	1437.6	1741.3	2127.3	2261.6	2414.7	1984.5	1531	997	574.9	428 4
Lat. 48°13′ N.	$\bar{K}_t$	0 621	0 678	0.672	0.597	0 611	0.602	0 666	0.630	0.629	0.593	0 516	0 548
El. 2277 ft.	t_0	13.3	17.3	31.1	47.8	59.3	67.3	76	73.2	61.2	49.2	31.0	18 6
Grand Junction, Colorado	$\bar{H}$	848	1210 7	1622 9	2002 2	2300 3	2645.4	2517 7	2157.2	1957 5	1394 8	969 7	793 4
Lat. 39°07′ N.	$\bar{K}_t$	0 597	0 633	0 643	0.632	0.643	0.704	0.690	0 65	0.705	0 654	0.59	0 621
El. 4849 ft	t_0	26 9	35 0	44.6	55 8	66.3	75.7	82.5	79.6	71.4	58.3	42.0	31.4
Grand Lake, Colo.	$\bar{H}$	735	1135 4	1579.3	1876.7	1974 9	2369.7	2103.3	1708.5	1715.8	1212 2	775.6	660 5
Lat. 40°15′ N.	$\bar{K}_t$	0 541	0.615	0 637	0 597	0.553	0.63	0.572	0.516	0 626	0.583	0 494	0 542
El. 8389 ft	t_0	18.5	23.1	28.5	39.1	48.7	56.6	62.8	61.5	55.5	45.2	30.3	22 6
Great Falls, Mont.	$\bar{H}$	524	869 4	1369.7	1621.4	1970.8	2179.3	2383	1986.3	1536.5	984.9	575 3	420.7
Lat 47°29′ N.	$\bar{K}_t$	0.552	0.596	0 631	0.551	0.565	0.580	0.656	0.627	0.626	0.574	0.503	0 518
El. 3664 ft	t_0	25.4	27.6	35.6	47.7	57.5	64.3	73.8	71.3	60.6	51 4	38.0	29.1
Greensboro, N. C.	$\bar{H}$	743 9	1031 7	1323 2	1755 3	1988.5	2111.4	2033 9	1810 3	1517.3	1202.6	908.1	690 8
Lat. 36°05′ N.	$\bar{K}_t$	0.469	0.499	0.409	0.543	0.554	0.563	0.552	0.538	0 527	0.531	0.501	0.479
El. 891 ft	t_0	42.0	44.2	51.7	60.8	69.9	78.0	80.2	78.9	73.9	62.7	51.5	43.2
Griffin, Georgia	$\bar{H}$	889.6	1135 8	1450.9	1923.6	2163 1	2176	2064.9	1961.2	1605.9	1352.4	1073.8	781.5
Lat. 33°15′ N.	$\bar{K}_t$	0.513	0.517	0.528	0.586	0.601	0.583	0.562	0.578	0.543	0.565	0.545	0.487
El. 980 ft	t_0	48.9	51.0	59.1	66.7	74.6	81.2	83.0	82.2	78.4	68	57.3	49.4
Hatteras, N. C.	$\bar{H}$	891.9	1184.1	1590.4	2128	2376.4	2438	2334.3	2085.6	1758.3	1337.6	1053.5	798.1
Lat. 35°13′ N.	$\bar{K}_t$	0.546	0.563	0.593	0.655	0.661	0.652	0.634	0.619	0.605	0.58	0.566	0.535
El. 7 ft	t_0	49.9	49.5	54.7	61.5	69.9	77.2	80.0	79.8	76.7	67.9	59.1	51.3
Indianapolis, Ind.	$\bar{H}$	526.2	797.4	1184.1	1481.2	1828	2042	2039.5	1832.1	1513.3	1094.4	662.4	491.1
Lat. 39°44′ N.	$\bar{K}_t$	0.380	0.424	0.472	0.47	0.511	0.543	0.554	0.552	0.549	0.520	0.413	0.391
El. 793 ft	t_0	31.3	33.9	43.0	54.1	64.9	74.8	79.6	77.4	70.6	59.3	44.2	33.4
Inyokern, Calif.	$\bar{H}$	1148.7	1554.2	2136.9	2594.8	2925.4	3108.8	2908.8	2759.4	2409.2	1819.2	1370.1	1094.4
Lat. 35°39′ N.	$\bar{K}_t$	0.716	0.745	0.803	0.8	0.815	0.830	0.790	0.820	0.834	0.795	0.743	0.742
El. 2440 ft	t_0	47.3	53.9	59.1	65.6	73.5	80.7	87.5	84.9	78.6	68.7	57.3	48.9
Ithaca, N. Y.	$\bar{H}$	434.3	755	1074.9	1322.9	1779.3	2025.8	2031.3	1736.9	1320.3	918.4	466.4	370.8
Lat. 42°27′ N.	$\bar{K}_t$	0.351	0.435	0.45	0.428	0.502	0.538	0.554	0.530	0.497	0.465	0.324	0.337
El. 950 ft	t_0	27.2	26.5	36	48.4	59.6	68.9	73.9	71.9	34.2	53.6	41.5	29.6
Lake Charles, La.	$\bar{H}$	899.2	1145.7	1487.4	1801.8	2080.4	2213.3	1968.6	1910.3	1678.2	1505.5	1122.1	875.6
Lat. 30°13′ N.	$\bar{K}_t$	0.473	0.492	0.521	0.542	0.578	0.597	0.538	0.558	0.553	0.597	0.524	0 494
El. 12 ft	t_0	55.3	58.7	63.5	70.9	77.4	83.4	84.8	85.0	81.5	73.8	62.6	56.0
Lander, Wyo.	$\bar{H}$	786.3	1146.1	1638	1988.5	2114	2492.2	2438.4	2120.6	1712.9	1301.8	837.3	694.8
Lat. 42°48′ N.	$\bar{K}_t$	0.65	0.672	0.691	0.647	0.597	0.662	0.665	0.649	0.647	0.666	0.589	0.643
El. 5370 ft	t_0	20.2	26.3	34.7	45.5	56.0	65.4	74.6	72.5	61.4	48.3	33.4	23.8

Table 3-1 (continued)

		Jan	Feb	Mar	Apr	May	Jun	July	Aug	Sep.	Oct	Nov	Dec
Las Vegas, Nev.	$\bar{H}$	1035.8	1438	1926.5	2322.8	2629.5	2799.2	2524	2342	2062	1602.6	1190	964.2
Lat. 36°05′ N.	$\bar{K}_t$	0.654	0.697	0.728	0.719	0.732	0.746	0.685	0.697	0.716	0.704	0.657	0.668
El. 2162 ft	t_0	47.5	53.9	60.3	69.5	78.3	88.2	95.0	92.9	85.4	71.7	57.8	50.2
Lemont, Illinois	$\bar{H}$	(590)	879	1255.7	1481.5	1866	2041.7	1990.8	1836.9	1469.4	1015.5	(639)	(531)
Lat. 41°40′ N.	$\bar{K}_t$	(0.464)	0.496	0.520	0.477	0.525	0.542	0.542	0.559	0.547	0.506	(0.433)	(0.467
El. 595 ft	t_0	28.9	30.3	39.5	49.7	59.2	70.8	75.6	74.3	67.2	57.6	43.0	30.6
Lexington, Ky.	$\bar{H}$	—	—	—	1834.7	2171.2	—	2246.5	2064.9	1775.6	1315.8	—	681.5
Lat. 38°02′ N.	$\bar{K}_t$	—	—	—	0.575	0.606	—	0.610	0.619	0.631	0.604	—	0.513
El. 979 ft	t_0	36.5	38.8	47.4	57.8	67.5	76.2	79.8	78.2	72.8	61.2	47.6	38.5
Lincoln, Neb.	$\bar{H}$	712.5	955.7	1299.6	1587.8	1856.1	2040.6	2011.4	1902.6	1543.5	1215.8	773.4	643.2
Lat. 40°51′ N.	$\bar{K}_t$	0.542	0.528	0.532	0.507	0.522	0.542	0.547	0.577	0.568	0.596	0.508	0.545
El. 1189 ft	t_0	27.8	32.1	42.4	55.8	65.8	76.0	82.6	80.2	71.5	59.9	43.2	31.8
Little Rock, Ark.	$\bar{H}$	704.4	974.2	1335.8	1669.4	1960.1	2091.5	2081.2	1938.7	1640.6	1282.6	913.6	701.1
Lat. 34°44′ N.	$\bar{K}_t$	0.424	0.458	0.496	0.513	0.545	0.559	0.566	0.574	0.561	0.552	0.484	0.463
El. 265 ft	t_0	44.6	48.5	56.0	65.8	73.1	76.7	85.1	84.6	78.3	67.9	54.7	46.7
Los Angeles, Calif. (WBAS)	$\bar{H}$	930.6	1284.1	1729.5	1948	2196.7	2272.3	2413.6	2155.3	1898.1	1372.7	1082.3	901.1
Lat. 33°56′ N.	$\bar{K}_t$	0.547	0.596	0.635	0.595	0.610	0.608	0.657	0.635	0.641	0.574	0.551	0.566
El. 99	t_0	56.2	56.9	59.2	61.4	64.2	66.7	69.6	70.2	69.1	66.1	62.6	58.7
Los Angeles, Calif. (WBO)	$\bar{H}$	911.8	1223.6	1640.9	1866.8	2061.2	2259	2428.4	2198.9	1891.5	1362.3	1053.1	877.8
Lat. 34°03′ N.	$\bar{K}_t$	0.538	0.568	0.602	0.571	0.573	0.605	0.66	0.648	0.643	0.578	0.548	0.566
	t_0	57.9	59.2	61.8	64.3	67.6	70.7	75.8	76.1	74.2	69.6	65.4	60.2
Madison, Wis.	$\bar{H}$	564.6	812.2	1232.1	1455.3	1745.4	2031.7	2046.5	1740.2	1443.9	993	555.7	495.9
Lat. 43°08′ N.	$\bar{K}_t$	0.49	0.478	0.522	0.474	0.493	0.540	0.559	0.534	0.549	0.510	0.396	0.467
El. 866 ft	t_0	21.8	24.6	35.3	49.0	61.0	70.9	76.8	74.4	65.6	53.7	37.8	25.4
Matanuska, Alaska	$\bar{H}$	119.2	345	—	1327.6	1628.4	1727.6	1526.9	1169	737.3	373.8	142.8	56.4
Lat. 61°30′ N.	$\bar{K}_t$	0.513	0.503	—	0.545	0.494	0.466	0.434	0.419	0.401	0.390	0.372	0.364
El. 180 ft	t_0	[illegible]	21.0	27.4	38.6	50.3	57.6	60.1	58.1	50.2	37.7	22.9	13.9
Medford, Oregon	$\bar{H}$	435.4	804.4	1259.8	1807.4	2216.2	2440.5	2607.4	2261.6	1672.3	1043.5	558.7	346.5
Lat. 42°23′ N.	$\bar{K}_t$	0.353	0.464	0.527	0.584	0.625	0.648	0.710	0.689	0.628	0.526	0.384	0.313
El. 1329 ft	t_0	39.4	45.4	50.8	56.3	63.1	69.4	76.9	76.4	69.6	58.7	47.1	40.5
Miami, Florida	$\bar{H}$	1292.2	1554.6	1828.8	2020.6	2068.6	1991.5	1992.6	1890.8	1646.8	1436.5	1321	1183.4
Lat. 25°47′ N.	$\bar{K}_t$	0.604	0.616	0.612	0.600	0.578	0.545	0.552	0.549	0.525	0.534	0.559	0.588
El. 9 ft	t_0	71.6	72.0	73.8	77.0	79.9	82.9	84.1	84.5	83.3	80.2	75.6	72.6
Midland, Texas	$\bar{H}$	1066.4	1345.7	1784.8	2036.1	2301.1	2317.7	2301.8	2193	1921.8	1470.8	1244.3	1023.2
Lat. 31°56′ N.	$\bar{K}_t$	0.587	0.596	0.638	0.617	0.639	0.622	0.628	0.643	0.642	0.600	0.609	0.611
El. 2854 ft	t_0	47.9	52.8	60.0	68.8	77.2	83.9	85.7	85.0	78.9	70.3	56.6	49.1
Nashville, Tenn.	$\bar{H}$	589.7	907	1246.8	1662.3	1997	2149.4	2079.7	1862.7	1600.7	1223.6	823.2	614.4
Lat. 36°07′ N.	$\bar{K}_t$	0.373	0.440	0.472	0.514	0.556	0.573	0.565	0.554	0.556	0.5[illegible]	0.454	0.426
El. 605 ft	t_0	42.6	45.1	52.9	63.0	71.4	80.1	83.2	81.9	76.6	65.4	52.3	44.3
Newport, R. I.	$\bar{H}$	565.7	856.4	1231.7	1484.8	1849	2019.2	1942.8	1687.1	1411.4	1035.4	656.1	527.7
Lat. 41°29′ N.	$\bar{K}_t$	0.438	0.482	0.507	0.477	0.520	0.536	0.529	0.513	0.524	0.512	0.44	0.460
El. 60 ft	t_0	29.5	32.0	39.6	48.2	58.6	67.0	73.2	72.3	66.7	56.2	46.5	34.4
New York, N. Y.	$\bar{H}$	539.5	790.8	1180.4	1426.2	1738.4	1994.1	1938.7	1605.9	1349.4	977.8	598.1	476
Lat. 40°46′ N.	$\bar{K}_t$	0.406	0.435	0.480	0.455	0.488	0.53	0.528	0.486	0.500	0.475	0.397	0.403
El. 52 ft	t_0	35.0	34.9	43.1	52.3	63.3	72.2	76.9	75.3	69.5	59.3	48.3	37.7
Oak Ridge, Tenn.	$\bar{H}$	604	895.9	1241.7	1689.6	1942.8	2068.4	1972.3	1795.6	1559.8	1194.8	796.3	610
Lat. 36°01′ N.	$\bar{K}_t$	0.382	0.435	0.471	0.524	0.541	0.551	0.536	0.534	0.542	0.527	0.438	0.422
El. 905 ft	t_0	41.9	44.2	51.7	61.4	69.8	77.8	80.2	78.8	74.5	62.7	50.4	42.5
Oklahoma City, Oklahoma	$\bar{H}$	938	1192.6	1534.3	1849.4	2005.1	2355	2273.8	2211	1819.2	1409.6	1085.6	897.4
Lat. 35°24′ N.	$\bar{K}_t$	0.580	0.571	0.576	0.570	0.558	0.629	0.618	0.656	0.628	0.614	0.588	0.608
El. 1304 ft	t_0	40.1	46.0	53.2	63.6	71.2	80.6	85.5	85.4	77.4	66.5	52.2	43.1
Ottawa, Ontario	$\bar{H}$	539.1	852.4	1250.5	1506.6	1857.2	2084.5	2045.4	1752.4	1326.6	826.9	458.7	408.5
Lat. 45°20′ N.	$\bar{K}_t$	0.499	0.540	0.554	0.502	0.529	0.554	0.560	0.546	0.521	0.450	0.359	0.436
El. 339 ft	t_0	14.6	15.6	27.7	43.8	57.5	67.5	71.9	69.8	61.5	48.9	35	19.6
Phoenix, Ariz.	$\bar{H}$	1126.6	1514.7	1967.1	2388.2	2709.6	2781.5	2450.5	2299.6	2131.3	1688.9	1290	1040.9
Lat. 33°26′ N.	$\bar{K}_t$	0.65	0.691	0.716	0.728	0.753	0.745	0.667	0.677	0.722	0.708	0.657	0.652
El. 1112 ft	t_0	54.2	58.8	64.7	72.2	80.8	89.2	94.6	92.5	87.4	75.8	63.6	56.7
[illegible]tland, Maine	$\bar{H}$	565.7	874.5	1329.5	1528.4	1923.2	2017.3	2095.6	1799.2	1428.8	1035	591.5	507.7
Lat. 43°39′ N.	$\bar{K}_t$	0.482	0.524	0.569	0.500	0.544	0.536	0.572	0.554	0.546	0.539	0.431	0.491
El. 63 ft	t_0	23.7	24.5	34.4	44.8	55.4	65.1	71.1	69.7	61.9	51.8	40.3	28.0

Table 3-1 (continued)

		Jan	Feb	Mar	Apr	May	Jun	July	Aug	Sep	Oct	Nov	Dec
Rapid City, S. D.	$\bar{H}$	687.8	1032 5	1503.7	1807	2028	2193 7	2235 8	2019 9	1628	1179.3	763 1	590.4
Lat. 44°09′ N.	$\bar{K}_t$	0.601	0.627	0.649	0.594	0.574	0 583	0 612	0 622	0.628	0.624	0 566	0.588
El. 3218 ft	t_0	24.7	27.4	34.7	48.2	58.3	67.3	76.3	75.0	64.7	52.9	38.7	29.2
Riverside, Calif.	$\bar{H}$	999.6	1335	1750.5	1933 2	2282.3	2492 6	2443 5	2263.8	1955.3	1509.6	1169	979.7
Lat. 33°57′ N.	$\bar{K}_t$	0 589	0.617	0.643	0 594	0.635	0 667	0.665	0 668	0.665	0 639	0.606	0.626
El. 1020 ft	t_0	55.3	57.0	60.6	65.0	69.4	74 0	81.0	81.0	78.5	71.0	63.1	57.2
Saint Cloud, Minn.	$\bar{H}$	632.8	976 7	1383	1598.1	1859.4	2003.3	2087 8	1828.4	1369 4	890.4	545.4	463.1
Lat. 45°35′ N.	$\bar{K}_t$	0 595	0.629	0 614	0.534	0 530	0.533	0 573	0 570	0.539	0.490	0.435	0.504
El. 1034 ft	t_0	13.6	16 9	29.8	46.2	58.8	68.5	74.4	71 9	62.5	50.2	32.1	18.3
Salt Lake City, Utah	$\bar{H}$	622 1	986	1301.1	1813.3	—	—	—	—	1689.3	1250.2	—	552 8
Lat. 40°46′ N.	$\bar{K}_t$	0.468	0.909	0 529	0 578	—	—	—	—	0 621	0 610	—	0.467
El 4227 ft	t_0	29.4	30.2	44.4	53.9	63.1	71.7	81.3	79.0	68.7	57.0	42 5	34.0
San Antonio, Tex	$\bar{H}$	1045	1299 2	1560 1	1664 6	2024.7	814.8	2364 2	2185 2	1844 6	1487 4	1104 4	954.6
Lat. 29°32′ N.	$\bar{K}_t$	0 541	0.550	0.542	0.500	0 563	0.220	0.647	0 637	0 603	0 584	0 507	0.528
El. 794 ft	t_0	53.7	58.4	65.0	72.2	79.2	85.0	87.4	87.8	82.6	74.7	63.3	56.5
Santa Maria, Calif.	$\bar{H}$	983.8	1296.3	1805 9	2067 9	2375 6	2599.6	2540.6	2293 3	1965 7	1566 4	1169	943 9
Lat. 34°54′ N.	$\bar{K}_t$	0 595	0.643	0.671	0.636	0.661	0.695	0 690	0 678	0 674	0 676	0.624	0.627
El. 238 ft	t_0	54.1	55.3	57.6	59.5	61.2	63.5	65.3	65.7	65 9	64.1	60.8	56.1
Sault Ste. Marie, Michigan	$\bar{H}$	488.6	843 9	1336.5	1559.4	1962.3	2064 2	2149.4	1767.9	1207	809 2	392 2	359 8
Lat. 46°28′ N.	$\bar{K}_t$	0 490	0 560	0 606	0.526	0.560	0.549	0 590	0.554	0.481	0 457	0 323	0.408
El. 724 ft	t_0	16.3	16.2	25.6	39.5	52.1	61.6	67.3	66.0	57.9	46.8	33.4	21.9
Sayville, N. Y.	$\bar{H}$	602.9	936 2	1259.4	1560 5	1857 2	2123 2	2040 9	1734.7	1446.8	1087.4	697 8	533 9
Lat 40°30′ N.	$\bar{K}_t$	0 453	0.511	0 510	0.498	0 522	0 564	0 555	0 525	0.530	0 527	0.450	0.447
El. 20 ft	t_0	35	34.9	43 1	52.3	63.3	72.2	76.9	75.3	69.5	59.3	48.3	37.7
Schenectady, N. Y.	$\bar{H}$	488.2	753 5	1026.6	1272 3	1553 1	1687.8	1662.3	1494.8	1124.7	820 6	436 2	356.8
Lat 42°50′ N.	$\bar{K}_t$	0.406	0 441	0.433	0.413	0 438	0.448	0 454	0.458	0.426	0 420	0.309	0.331
El 217 ft	t_0	24 7	24.6	34.9	48.3	61.7	70.8	76.9	73.7	64.6	53.1	40.1	28.0
Seattle, Wash.	$\bar{H}$	282.6	520 6	992 2	1507	1881.5	1909.9	2110.7	1688.5	1211.8	702.2	386.3	239 5
Lat. 47°27′ N.	$\bar{K}_t$	0.296	0.355	0.456	0.510	0 538	0.508	0.581	0.533	0.492	0 407	0 336	0.292
El. 386 ft	t_0	42.1	45.0	48.9	54.1	59.8	64.4	68.4	67.9	63.3	56.3	48.4	44.4
Seattle, Wash.	$\bar{H}$	252	471.6	917.3	1375 6	1664.9	1724	1805.1	1617	1129.1	638	325.5	218.1
Lat 47°36′ N.	$\bar{K}_t$	0.266	0.324	0.423	0.468	0.477	0.459	0.498	0.511	0.459	0.372	0 284	0.269
El. 14 ft	t_0	38 9	42.9	46.9	51.9	58.1	62.8	67.2	66.7	61 6	54.0	45.7	41.5
Seabrook, N. J.	$\bar{H}$	591.9	854.2	1195.6	1518.8	1800.7	1964.6	1949.8	1715	1445.7	1071.9	721.8	522 5
Lat. 39°30′ N.	$\bar{K}_t$	0.426	[illegible]	0.476	0.481	0.504	0.522	0.530	0.517	0.524	0.508	0.449	0.416
El. 100 ft	t_0	39.5	[illegible]	43.9	54.7	64.9	74.1	79.8	77.7	69.7	61.2	48.5	39.3
Spokane, Wash.	$\bar{H}$	446.1	837	1200	1764.6	2104 4	2226.5	2479.7	2076	1511	844.6	486 3	279
Lat. 47°40′ N.	$\bar{K}_t$	0 478	0.579	0.556	0.602	0 603	0.593	0 684	0.656	0.616	0.494	0.428	0.345
El. 1968	t_0	26.5	31.7	40.5	49.2	57.9	64.6	73.4	71.7	62.7	51.5	37.4	30.5
State College, Pa.	H	501.8	749.1	1106.6	1399.2	1754.6	2027.6	1968.2	1690	1336.1	1017	580.1	443.9
Lat. 40°48′ N.	$\bar{K}_t$	0.381	0.413	0 451	0.448	0.493	0.539	0.536	0.512	0.492	0.496	0.379	0.376
El. 1175 ft	t_0	31.3	31.4	39.8	51.3	63.4	71.8	75.8	73.4	66.1	55.6	43.2	32.6
Stillwater, Okla.	$\bar{H}$	763.8	1081.5	1463.8	1702.6	1879.3	2235.8	2224.3	2039.1	1724.3	1314	991.5	783
Lat. 36°09′ N.	$\bar{K}_t$	0.484	0.527	0 555	0.528	0.523	0.596	0.604	0.607	0.599	0.581	0.548	0.544
El. 910 ft	t_0	41.2	45.6	53.8	64.2	71.6	81.1	85.9	85.9	77.5	67.6	52.6	43.9
Tampa, Fla.	$\bar{H}$	1223.6	1461.2	1771.9	2016.2	2228	2146.5	1991.9	1845.4	1687.8	1493.3	1328.4	1119.5
Lat. 27°55′ N.	$\bar{K}_t$	0.605	0.600	0.606	0.602	0.620	0.583	0.548	0.537	0.546	0.572	0.590	0.589
El. 11 ft	t_0	64.2	65.7	68.8	74.3	79.4	83.0	84.0	84.4	82.9	77.2	69.6	65.5
Toronto, Ontario	$\bar{H}$	451.3	674.5	1088.9	1388.2	1785.2	1941.7	1968.6	1622.5	1284.1	835	458.3	352.8
Lat. 43°41′ N.	$\bar{K}_t$	0.388	0.406	0.467	0.455	0.506	0.516	0.539	0.500	0.493	0.438	0.336	0.346
El. 379 ft	t_0	26.5	26.0	34.2	[illegible]	58	68.4	73.8	71.8	64.3	52.6	40.9	30.2
Tucson, Arizona	$\bar{H}$	1171.9	1453.8	—	2434.7	—	2601.4	2292.2	2179.7	2122.5	1640.9	1322.1	1132.1
Lat. 32°07′ N.	$\bar{K}_t$	0.648	0.646	—	0.738	—	0.698	0.625	0.640	0.710	0 672	0.650	0.679
El. 2556 ft	t_0	53.7	57.3	62.3	69.7	78.0	87.0	90.1	87.4	84.0	73.9	62.5	56.1
Upton, N. Y.	$\bar{H}$	583	872.7	1280.4	1609.9	1891.5	2159	2044.6	1789.6	1472.7	1102.6	686.7	551.3
Lat. 40°52′ N.	$\bar{K}_t$	0.444	0.483	0.522	0.514	0.532	0.574	0.557	0.542	0.542	0.538	0.448	0.467
El. 75 ft	t_0	35.0	34.9	43.1	52.3	63.3	72.2	76.9	75.3	69.5	59.3	48.3	37.7
Washington, D. C. (WBCO)	$\bar{H}$	632 4	901.5	1255	1600.4	1846.8	2080.8	1929.9	1712 2	1446.1	1083.4	763.5	594.1
Lat. 38°51′ N.	$\bar{K}_t$	0.445	0.470	0.496	0.504	0.516	0.553	0.524	0.516	0.520	0.406	0.464	0.460
El. 64 ft	t_0	38.4	39.6	48.1	57.5	67.7	76.2	79.9	77.9	72.2	60.9	50.2	40.2
Winnipeg, Man.	$\bar{H}$	488.2	835.4	1354.2	1641.8	1904.4	1962	2123.6	1761.2	1190.4	767.5	444.6	345
Lat. 49°54′ N.	$\bar{K}_t$	0.601	0.630	0.661	0.574	0.550	0.524	0.587	0.567	0.504	0.482	0.436	0.503
El. 786 ft	t_0	3.2	7.1	21.3	40.9	55.9	65.3	71.9	69.4	58.6	45.6	25.2	10.1

Table 3-2

Monthly Average Daily Extraterrestrial Radiation, $\overline{H}_o$

Btu/ft^2 and kJ/m^2

LATITUDE	JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC
20	2349.	2676.	3024.	3307.	3428.	3451.	3425.	3338.	3112.	2768.	2425.	2250.
	26644.	30359.	34307.	37515.	38884.	39144.	38893.	37864.	35300.	31402.	27512.	25519.
25	2107.	2478.	2896.	3271.	3496.	3530.	3491.	3335.	3018.	2593.	2196.	1998.
	23902.	28115.	32848.	37111.	39356.	40046.	39606.	37832.	34238.	29413.	24909.	22669.
30	1854.	2264.	2745.	3212.	3488.	3588.	3532.	3307.	2902.	2399.	1953.	1738.
	21034.	25679.	31141.	36436.	39569.	40706.	40071.	37534.	32917.	27213.	22161.	19714.
35	1593.	2034.	2574.	3129.	3489.	3625.	3551.	3259.	2763.	2188.	1701.	1471.
	18069.	23072.	29200.	35497.	39530.	41129.	40292.	36976.	31348.	24820.	19296.	16687.
40	1326.	1791.	2384.	3024.	3460.	3643.	3551.	3188.	2604.	1962.	1441.	1201.
	15043.	20319.	27040.	34303.	39247.	41328.	40281.	36166.	29542.	22255.	16344.	13626.
45	1058.	1538.	2175.	2897.	3415.	3643.	3531.	3096.	2425.	1723.	1176.	933.
	11998.	17448.	24677.	32869.	38737.	41322.	40055.	35118.	27515.	19541.	13344.	10579.
50	792.	1277.	1951.	2751.	3352.	3627.	3495.	2984.	2229.	1472.	912.	670.
	8987.	14490.	22131.	31209.	38025.	41147.	39644.	33851.	25283.	16705.	10342.	7605.
55	536.	1013.	1712.	2587.	3275.	3602.	3447.	2855.	2015.	1214.	652.	422.
	6082.	11486.	19423.	29345.	37152.	40863.	39100.	32391.	22863.	13778.	7396.	4791.
60	299.	748.	1461.	2407.	3190.	3578.	3395.	2713.	1787.	952.	405.	201.
	3395.	8486.	16576.	27308.	36188.	40585.	38513.	30779.	20277.	10798.	4598.	2277.

Table 3-3

$\bar{R}$ for $\bar{K}_T$ = .30

LATITUDE	JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC
Tilt = Latitude - 15°												
20	1.06	1.05	1.03	1.01	1.01	1.00	1.01	1.01	1.02	1.04	1.06	1.07
25	1.12	1.08	1.05	1.02	1.00	.99	1.00	1.01	1.03	1.07	1.11	1.09
30	1.14	1.13	1.07	1.03	1.00	.98	.99	1.02	1.05	1.11	1.12	1.16
35	1.23	1.19	1.10	1.04	.99	.97	.98	1.02	1.07	1.15	1.20	1.26
40	1.35	1.28	1.14	1.05	.99	.97	.98	1.02	1.09	1.22	1.31	1.[illegible]
45	1.53	1.31	1.18	1.07	1.00	.97	.98	1.03	1.13	1.31	1.46	1.68
50	1.71	1.45	1.25	1.09	1.00	.97	.98	1.04	1.17	1.36	1.73	1.88
55	2.27	1.67	1.34	1.11	1.01	.96	.98	1.06	1.23	1.52	1.99	2.74
60	3.09	2.07	1.47	1.15	1.01	.96	.98	1.09	1.31	1.77	2.92	4.41
Tilt = Latitude												
20	1.15	1.10	1.04	.99	.96	.94	.95	.98	1.02	1.08	1.14	1.17
25	1.22	1.13	1.06	.99	.95	.93	.94	.98	1.03	1.11	1.19	1.18
30	1.23	1.18	1.08	1.00	.94	.92	.93	.97	1.04	1.14	1.20	1.26
35	1.33	1.25	1.10	1.00	.94	.90	.92	.97	1.06	1.19	1.29	1.38
40	1.47	1.36	1.14	1.01	.93	.90	.91	.97	1.08	1.26	1.41	1.54
45	1.68	1.38	1.19	1.02	.93	.89	.91	.98	1.11	1.36	1.58	1.88
50	1.88	1.53	1.26	1.04	.93	.88	.90	.98	1.15	1.41	1.89	2.10
55	2.53	1.77	1.35	1.06	.93	.87	.90	1.00	1.21	1.58	2.18	3.12
60	3.48	2.21	1.48	1.09	.93	.87	.89	1.02	1.29	1.85	3.26	5.07

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Table 3-3 (continued)

$\overline{R}$ for $\overline{K}_T = .30$

Tilt = Latitude + 15°

LATITUDE	JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC
20	1.19	1.10	1.01	.92	.88	.85	.86	.91	.98	1.07	1.16	1.22
25	1.27	1.13	1.02	.93	.87	.84	.85	.90	.98	1.09	1.22	1.22
30	1.26	1.18	1.04	.93	.85	.82	.83	.89	.99	1.13	1.22	1.31
35	1.37	1.25	1.06	.93	.84	.80	.82	.89	1.00	1.18	1.31	1.43
40	1.51	1.37	1.09	.93	.83	.80	.81	.89	1.02	1.25	1.43	1.61
45	1.74	1.37	1.14	.94	.83	.78	.80	.89	1.05	1.35	1.62	1.99
50	1.95	1.53	1.20	.95	.82	.77	.79	.89	1.08	1.38	1.95	2.21
55	2.65	1.78	1.29	.97	.82	.76	.78	.90	1.14	1.56	2.25	3.32
60	3.66	2.23	1.42	.99	.81	.75	.77	.91	1.21	1.84	3.39	5.41

Vertical

LATITUDE	JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC
20	.88	.72	.58	.44	.39	.38	.38	.42	.52	.67	.83	.93
25	1.00	.79	.63	.49	.41	.39	.40	.45	.57	.73	.93	.95
30	1.01	.88	.68	.53	.44	.41	.42	.49	.61	.80	.95	1.07
35	1.14	.98	.74	.57	.47	.44	.45	.53	.66	.89	1.07	1.22
40	1.32	1.13	.81	.62	.51	.47	.49	.56	.72	.99	1.23	1.44
45	1.59	1.17	.89	.66	.55	.50	.52	.61	.79	1.13	1.45	1.87
50	1.83	1.36	.99	.72	.59	.54	.56	.65	.87	1.20	1.82	2.10
55	2.56	1.64	1.12	.78	.62	.57	.59	.71	.96	1.41	2.14	3.27

Table 3-4

$\overline{R}$ for $\overline{K}_T = .40$

Tilt = Latitude - 15°

LATITUDE	JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC
20	1.07	1.05	1.03	1.01	1.01	1.00	1.00	1.01	1.02	1.05	1.07	1.08
25	1.15	1.10	1.05	1.02	1.00	.99	.99	1.01	1.04	1.08	1.13	1.11
30	1.18	1.16	1.08	1.03	1.00	.98	.98	1.02	1.06	1.13	1.16	1.20
35	1.29	1.23	1.12	1.05	.99	.97	.98	1.02	1.09	1.19	1.26	1.33
40	1.44	1.35	1.17	1.06	.99	.97	.98	1.03	1.12	1.27	1.39	1.51
45	1.68	1.40	1.24	1.09	1.00	.97	.98	1.05	1.17	1.30	1.58	1.85
50	1.90	1.58	1.32	1.12	1.01	.97	.99	1.06	1.23	1.46	1.92	2.11
55	2.60	1.86	1.44	1.16	1.02	.97	.99	1.09	1.30	1.66	2.25	3.16
60	3.59	2.36	1.61	1.21	1.03	.97	1.00	1.13	1.41	1.99	3.40	5.16

Tilt = Latitude

LATITUDE	JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC
20	1.19	1.12	1.05	.99	.95	.93	.94	.97	1.03	1.10	1.17	1.21
25	1.28	1.17	1.08	1.00	.94	.92	.93	.97	1.04	1.14	1.24	1.23
30	1.30	1.23	1.11	1.00	.94	.91	.92	.97	1.06	1.19	1.26	1.34
35	1.43	1.32	1.14	1.01	.93	.89	.91	.98	1.09	1.25	1.37	1.49
40	1.61	1.45	1.19	1.03	.93	.89	.91	.98	1.12	1.34	1.53	1.70
45	1.88	1.49	1.26	1.05	.93	.89	.91	.99	1.16	1.47	1.75	2.12
50	2.13	1.69	1.35	1.08	.94	.88	.90	1.01	1.22	1.54	2.14	2.40
55	2.94	2.00	1.47	1.11	.94	.88	.90	1.03	1.30	1.76	2.50	3.66
60	4.09	2.56	1.64	1.16	.95	.88	.91	1.06	1.41	2.11	3.84	5.98

Table 3-4 (continued)

$\overline{R}$ for $\overline{K}_T$ = .40

LATITUDE	JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC
Tilt = Latitude + 15°												
20	1.25	1.14	1.03	.92	.86	.83	.84	.90	.98	1.10	1.22	1.29
25	1.35	1.19	1.05	.93	.85	.81	.83	.89	1.00	1.14	1.30	1.30
30	1.36	1.25	1.07	.93	.84	.80	.82	.89	1.01	1.19	1.30	1.41
35	1.49	1.34	1.11	.94	.83	.79	.81	.89	1.03	1.25	1.42	1.57
40	1.68	1.48	1.15	.95	.83	.78	.80	.89	1.06	1.34	1.58	1.80
45	1.98	1.51	1.22	.96	.83	.77	.80	.90	1.10	1.47	1.82	2.27
50	2.24	1.72	1.30	.99	.83	.76	.79	.91	1.16	1.53	2.24	2.56
55	3.11	2.03	1.42	1.01	.83	.76	.79	.93	1.23	1.76	2.61	3.92
60	4.32	2.61	1.59	1.05	.83	.75	.78	.95	1.33	2.11	4.03	6.42
Vertical												
20	.95	.76	.57	.40	.34	.33	.33	.37	.50	.69	.89	1.01
25	1.09	.84	.63	.46	.36	.34	.35	.41	.56	.77	1.01	1.04
30	1.11	.94	.70	.50	.40	.36	.38	.46	.61	.85	1.04	1.18
35	1.28	1.07	.77	.56	.44	.40	.41	.50	.68	.95	1.19	1.38
40	1.51	1.25	.86	.61	.48	.44	.46	.55	.75	1.08	1.39	1.65
45	1.84	1.30	.96	.67	.53	.48	.50	.60	.83	1.25	1.66	2.17
50	2.13	1.54	1.08	.74	.58	.52	.54	.66	.93	1.34	2.12	2.47
55	3.04	1.90	1.24	.82	.62	.55	.58	.73	1.04	1.60	2.51	3.90

Table 3-5

$\overline{R}$ for $\overline{K}_T = .50$

LATITUDE	JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT.	NOV	DEC
Tilt = Latitude - 15°												
20	1.08	1.06	1.03	1.01	1.01	1.00	1.00	1.01	1.02	1.05	1.08	1.09
25	1.17	1.11	1.06	1.03	1.00	.98	.99	1.01	1.04	1.09	1.15	1.13
30	1.21	1.18	1.10	1.04	.99	.97	.98	1.02	1.07	1.15	1.19	1.24
25	1.34	1.27	1.14	1.05	.99	.97	.98	1.03	1.10	1.22	1.30	1.38
40	1.52	1.40	1.20	1.08	1.00	.97	.98	1.04	1.14	1.32	1.46	1.60
45	1.80	1.47	1.28	1.11	1.01	.97	.98	1.06	1.20	1.45	1.69	1.99
50	2.06	1.68	1.38	1.14	1.02	.97	.99	1.08	1.27	1.54	2.08	2.30
55	2.87	2.01	1.52	1.19	1.03	.97	1.00	1.12	1.36	1.78	2.46	3.51
60	4.01	2.60	1.72	1.25	1.05	.98	1.01	1.16	1.49	2.16	3.80	5.78
Tilt = Latitude												
20	1.22	1.14	1.06	.99	.94	.92	.93	.97	1.03	1.12	1.20	1.25
25	1.33	1.20	1.09	1.00	.94	.91	.92	.97	1.05	1.16	1.29	1.28
30	1.36	1.28	1.13	1.01	.93	.89	.91	.97	1.08	1.22	1.31	1.40
35	1.51	1.38	1.18	1.02	.93	.89	.90	.98	1.11	1.30	1.45	1.58
40	1.72	1.53	1.24	1.04	.93	.88	.90	.99	1.15	1.41	1.63	1.83
45	2.05	1.59	1.32	1.07	.93	.88	.90	1.01	1.20	1.56	1.89	2.31
50	2.34	1.83	1.42	1.10	.94	.88	.91	1.03	1.28	1.64	2.35	2.65
55	3.28	2.19	1.57	1.15	.95	.88	.91	1.06	1.37	1.91	2.77	4.09
60	4.59	2.85	1.77	1.21	.97	.88	.92	1.09	1.50	2.32	4.31	6.74

Table 3-5 (continued)

$\overline{R}$ for $\overline{K}_T$ = .50

Tilt = Latitude + 15°

LATITUDE	JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC
20	1.30	1.17	1.04	.92	.85	.81	.82	.89	.99	1.12	1.26	1.34
25	1.42	1.23	1.07	.93	.84	.79	[illegible]	.89	1.01	1.17	1.36	1.36
30	1.43	1.31	1.10	.93	.83	.78	.80	.89	1.03	1.23	1.37	1.50
35	1.59	1.42	1.15	.95	.82	.77	.79	.89	1.06	1.31	1.51	1.69
40	1.82	1.58	1.20	.96	.82	.77	.79	.90	1.10	1.42	1.71	1.96
45	2.18	1.62	1.28	.98	.83	.76	.79	.91	1.15	1.57	1.99	2.50
50	2.48	1.87	1.38	1.01	.83	.76	.79	.93	1.21	1.65	2.48	2.84
55	3.49	2.25	1.52	1.05	.84	.75	.79	.95	1.30	1.92	2.91	4.42
60	4.88	2.93	1.73	1.10	.84	.75	.79	.98	1.43	2.34	4.56	7.26

Vertical

20	1.00	.78	.57	.37	.29	.29	.29	.33	.49	.71	.94	1.08
25	1.17	.88	.64	.43	.32	.29	.31	.38	.55	.79	1.07	1.11
30	1.19	1.00	.71	.49	.37	.32	.34	.43	.62	.89	1.11	1.28
35	1.39	1.15	.80	.55	.41	.36	.38	.48	.69	1.01	1.28	1.51
40	1.66	1.35	.90	.61	.46	.41	.43	.54	.77	1.16	1.51	1.83
45	2.05	1.42	1.01	.68	.52	.45	.48	.60	.86	1.35	1.83	2.42
50	2.37	1.70	1.16	.76	.57	.50	.53	.67	.97	1.46	2.37	2.77

Table 3-6

$\overline{R}$ for $\overline{K}_T$ = .60

LATITUDE	JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC
					Tilt = Latitude - 15°							
20	1.09	1.06	1.04	1.01	1.00	1.00	1.00	1.01	1.03	1.06	1.08	1.10
25	1.19	1.13	1.07	1.03	1.00	.98	.99	1.01	1.05	1.11	1.17	1.15
30	1.24	1.20	1.11	1.04	.99	.97	.98	1.02	1.08	1.17	1.21	1.27
35	1.39	1.31	1.17	1.06	.99	.96	.98	1.03	1.12	1.25	1.35	1.44
40	1.60	1.46	1.23	1.09	1.00	.96	.98	1.05	1.17	1.37	1.53	1.68
45	1.92	1.54	1.32	1.12	1.01	.97	.99	1.07	1.23	1.52	1.79	2.13
50	2.22	1.79	1.44	1.17	1.03	.97	.99	1.10	1.32	1.63	2.24	2.49
55	3.13	2.17	1.60	1.23	1.05	.98	1.01	1.14	1.42	1.90	2.68	3.85
60	4.41	2.83	1.83	1.30	1.07	.99	1.02	1.19	1.57	2.34	4.19	6.39
					Tilt = Latitude							
20	1.25	1.16	1.07	.99	.94	.91	.92	.97	1.04	1.13	1.23	1.28
25	1.37	1.23	1.11	1.00	.93	.89	.91	.97	1.06	1.19	1.33	1.32
30	1.41	1.32	1.15	1.01	.92	.88	.90	.97	1.09	1.26	1.37	1.47
35	1.59	1.44	1.21	1.03	.92	.88	.90	.98	1.13	1.35	1.52	1.67
40	1.84	1.61	1.28	1.06	.93	.88	.90	1.00	1.18	1.47	1.73	1.96
45	2.21	1.69	1.37	1.09	.94	.88	.90	1.02	1.25	1.65	2.03	2.50
50	2.54	1.96	1.50	1.13	.95	.88	.91	1.05	1.33	1.75	2.55	2.90
55	3.61	2.38	1.66	1.19	.96	.88	.92	1.08	1.44	2.05	3.04	4.53
60	5.08	3.13	1.90	1.26	.98	.89	.93	1.13	1.59	2.53	4.79	7.48

Table 3-6 (continued)

$\overline{R}$ for $\overline{K}_T$ = .60

LATITUDE	JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC
					Tilt = Latitude + 15°							
20	1.35	1.20	1.05	.92	.83	.79	.81	.88	1.00	1.15	1.30	1.40
25	1.48	1.27	1.09	.93	.82	.78	.80	.88	1.02	1.21	1.41	1.42
30	1.51	1.36	1.13	.94	[illegible]	.77	.79	.88	1.05	1.28	1.44	1.58
35	1.70	1.49	1.18	.95	.82	.76	.78	.89	1.08	1.37	1.60	1.80
40	1.96	1.68	1.25	.98	.82	.76	.78	.90	1.13	1.50	1.83	2.12
45	2.37	1.73	1.34	1.01	.83	.75	.78	.92	1.19	1.68	2.16	2.73
50	2.71	2.07	1.46	1.04	.83	.75	.79	.94	1.27	1.77	2.72	3.13
55	3.87	2.46	1.63	1.09	.85	.75	.79	.97	1.38	2.08	3.21	4.91
60	5.42	3.24	1.86	1.15	.86	.76	.80	1.01	1.52	2.56	5.08	8.08
						Vertical						
20	1.06	.81	.56	.34	.25	.25	.25	.29	.47	.72	.98	1.15
25	1.24	.92	.64	.41	.29	.25	.26	.35	.54	.82	1.14	1.18
30	1.27	1.05	.73	.47	.33	.29	.30	.41	.62	.93	1.18	1.38
35	1.50	1.22	.82	.54	.38	.33	.35	.47	.70	1.07	1.38	1.64
40	1.81	1.45	.94	.61	.44	.38	.40	.53	.79	1.23	1.64	2.00
45	2.26	1.53	1.07	.69	.50	.43	.46	.60	.90	1.45	2.01	2.66
50	2.62	1.85	1.23	.78	.56	.48	.52	.67	1.02	1.58	2.61	3.07

Table 3-7
Monthly Average Daily Radiation on a Tilted Surface
$\overline{H}_{TILT}$ for $\overline{K}_T = .3$, Tilt = Latitude - 15°, A = Btu/Ft², B = kJ/m²

LAT	JAN		FEB		MAR		APRIL		MAY		JUNE		JULY		AUG		SEPT		OCT		NOV		DEC	
	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B
20	746	8472	842	9563	933	10600	1001	11367	1037	11781	1034	11743	1038	11784	1010	11472	951	10801	863	9797	770	8749	721	8192
25	707	8031	802	9109	911	10347	1000	11355	1040	11807	1047	11894	1046	11882	1009	11463	931	10579	831	9442	730	8295	653	7412
30	633	7193	767	8705	880	9996	991	11259	1045	11870	1054	11968	1048	11901	1011	11485	913	10369	798	9062	656	7446	604	6860
35	587	6667	725	8237	848	9636	975	11075	1034	11740	1054	11968	1043	11845	996	11314	886	10062	654	8563	611	6946	555	6308
40	536	6092	687	7802	814	9248	951	10805	1026	11656	1058	12026	1043	11843	974	11067	851	9660	717	8145	566	6423	504	5723
45	485	5507	604	6857	769	8736	929	10551	1023	11621	1058	12024	1037	11776	956	10851	821	9326	676	7679	515	5845	470	5332
50	406	4610	555	6303	731	8299	899	10205	1004	11408	1054	11974	1026	11655	930	10562	781	8874	600	6816	472	5367	378	4289
55	365	4142	507	5754	688	7808	860	9772	991	11257	1036	11768	1012	11495	907	10300	743	8436	553	6283	389	4415	347	3938
60	301	3147	464	5270	644	7310	830	9421	965	10964	1029	11688	997	11323	886	10064	702	7969	505	5733	354	4027	265	3012

Table 3-8
Monthly Average Daily Radiation on a Tilted Surface
$\overline{H}_{TILT}$ for $\overline{K}_T = .3$, Tilt = Latitude, A = Btu/ft², B = kJ/m²

LAT	JAN		FEB		MAR		APRIL		MAY		JUNE		JULY		AUG		SEPT		OCT		NOV		DEC	
	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B
20	809	9192	882	10018	942	10703	981	11141	986	11199	972	11039	976	11084	980	11132	951	10802	896	10174	829	9409	789	8957
25	770	8748	839	9531	920	10445	970	11022	988	11216	984	11172	983	11168	979	11122	932	10579	862	9794	783	8892	706	8024
30	683	7761	800	9090	888	10090	962	10930	982	11158	989	11234	984	11180	962	10922	904	10270	819	9306	702	7978	656	7451
35	635	7209	762	8652	848	9636	938	10649	982	11147	978	11104	979	11120	948	10760	878	9969	780	8861	658	7468	608	6908
40	584	6633	730	8290	814	9248	915	10393	964	10950	982	11158	968	10997	927	10524	843	9571	741	8412	609	6918	554	6295
45	532	6047	636	7223	776	8809	886	10057	952	10808	971	11032	963	10935	909	10324	807	9162	702	7973	557	6325	525	5966
50	446	5068	586	6651	737	8366	857	9737	934	10608	956	10862	942	10703	876	9952	768	8723	622	7066	516	5864	422	4791
55	406	4616	537	6099	693	7866	822	9332	913	10365	939	10665	930	10557	856	9717	731	8299	575	6531	426	4837	395	4484
60	309	3513	495	5626	648	7360	786	8930	889	10096	933	10593	905	10283	829	9418	691	7847	527	5993	396	4497	305	3463

Table 3-9

Monthly Average Daily Radiation on a Tilted Surface

$\overline{H}_{TILT}$ for $\overline{K}_T$ = .3, Tilt = Latitude + 15°, A = Btu/ft^2, B = kJ/m^2

LAT	JAN		FEB		MAR		APRIL		MAY		JUNE		JULY		AUG		SEPT		OCT		NOV		DEC	
	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B
20	838	9512	882	10018	915	10395	912	10354	903	10265	879	9982	884	10034	910	10336	913	10378	888	10080	843	9574	822	9340
25	802	9107	839	9530	885	10051	912	10353	904	10272	889	10091	889	10099	890	10215	886	10066	847	9618	803	9117	731	8297
30	700	7951	800	9090	856	9716	895	10166	888	10090	882	10014	879	9978	883	10026	861	9776	812	9225	714	8110	682	7748
35	654	7426	762	8652	818	9286	872	9903	877	9962	869	9870	873	9912	870	9873	828	9404	774	8786	668	7583	630	7159
40	600	6814	735	8351	779	8842	8843	9570	860	9772	873	9918	862	9788	850	9656	796	9040	735	8345	617	7012	580	6581
45	551	6263	631	7171	743	8440	816	9269	849	9645	851	9669	825	9373	826	9377	763	8667	697	7914	571	6485	556	6316
50	463	5257	586	6651	701	7967	783	8894	824	9354	837	9505	827	9396	796	9038	721	8192	609	6916	533	6050	444	5042
55	426	4835	540	6134	662	7517	752	8539	805	9139	820	9317	806	9149	770	8745	688	7819	568	6448	440	4992	420	4772
60	328	3728	500	5677	622	7061	714	8110	774	8794	804	9131	783	8896	740	8402	648	7[illegible]60	525	5960	412	4676	325	3696

Table 3-10

Monthly Average Daily Radiation on a Tilted Surface

$\overline{H}_{TILT}$ for $\overline{K}_T$ = .3, Vertical, A = Btu/ft^2, B = kJ/m^2

LAT	JAN		FEB		MAR		APRIL		MAY		JUNE		JULY		AUG		SEPT		OCT		NOV		DEC	
	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B
20	619	7034	577	6557	525	5969	436	4951	401	4549	392	4462	390	4433	420	4770	485	5507	556	6311	603	6850	627	7120
25	631	7171	587	6663	547	6208	480	5455	426	4841	413	4685	418	4753	450	5107	516	5855	567	6441	612	6950	569	6461
30	561	6373	597	6780	559	6353	510	5793	460	5223	441	5007	445	5049	485	5517	530	6024	575	6531	556	6316	557	6328
35	544	6180	597	6783	571	6482	534	6070	491	5574	478	5429	479	5439	518	5879	547	6207	584	6627	545	6194	538	6107
40	524	5957	606	6888	578	6571	562	6380	528	6005	513	5827	521	5921	535	6076	562	6381	582	6610	530	6030	518	5886
45	504	57[illegible]	539	6124	850	6589	573	6508	563	6392	546	6198	550	6249	733	8323	574	6521	583	6624	511	5804	523	5934

Table 3-11
Monthly Average Daily Radiation on a Tilted Surface
$\overline{H}_{TILT}$ for $\overline{K}_T$ = .4, Tilt = Latitude - 15°, A = Btu/ft^2, B = kJ/m^2

LAT	JAN		FEB		MAR		APRIL		MAY		JUNE		JULY		AUG		SEPT		OCT		NOV		DEC	
	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B
20	1004	11403	1122	12750	1245	14134	1335	15156	1383	15709	1378	15658	1370	15557	1347	15297	1268	14402	1161	13189	1037	11775	971	11024
25	968	10995	1089	12371	1215	13796	1333	15141	1386	15742	1396	15858	1381	15684	1346	15284	1254	14243	1118	12706	991	11259	886	10065
30	874	9928	1049	11916	1185	13492	1322	15011	1394	15828	1405	15957	1397	15868	1349	15314	1229	13956	1083	12300	905	10282	833	9463
35	821	9324	999	11351	1152	13082	1313	14909	1378	15654	1405	15958	1391	15794	1328	15086	1203	13667	1040	11814	723	8213	782	8877
40	763	8665	966	10972	1114	12654	1281	14544	1369	15542	1412	16035	1388	15760	1312	14900	1165	13234	995	11305	800	9087	725	8230
45	710	8063	860	9771	1078	12240	1262	14330	1364	15495	1412	16032	1383	15701	1299	14749	1139	12877	957	10865	743	8433	684	7828
50	601	6830	806	9158	1029	11685	1231	13982	1353	15362	1406	15965	1382	15699	1264	14353	1095	12439	859	9758	699	7943	565	6419
55	557	6325	752	8546	985	11187	1199	13616	1335	15158	1396	15854	1363	15483	1244	14122	1047	11888	806	9148	586	6656	533	6055
60	429	4875	705	8010	940	10675	1164	13217	1313	14909	1387	15747	1357	15405	1225	13912	1007	11436	757	8595	551	6253	414	4700

Table 3-12
Monthly Average Daily Radiation on a Tilted Surface
$\overline{H}_{TILT}$ for $\overline{K}_T$ = .4, Tilt = Latitude, A = Btu/ft^2, B = kJ/m^2

LAT	JAN		FEB		MAR		APRIL		MAY		JUNE		JULY		AUG		SEPT		OCT		NOV		DEC	
	A	B	A	8	A	8	A	B	A	B	A	B	A	8	A	8	A	B	A	B	A	8	A	8
20	1117	12683	1198	13600	1269	14408	1308	14856	1301	14775	1282	14561	1288	14623	1294	14691	1281	14544	1217	13817	1134	12875	1088	12351
25	1078	12238	1159	13157	1250	14190	1307	14844	1303	14798	1298	14737	1297	14733	1292	14678	1254	14243	1181	13412	1088	12354	982	11153
30	963	10938	1113	12634	1215	13826	1283	14574	1310	14878	1305	14817	1298	14746	1282	14563	1229	13956	1208	13715	984	11169	930	10567
35	910	10335	1073	12182	1172	13315	1263	14341	1294	14705	1289	14642	1291	14666	1276	14494	1203	13667	1093	12410	931	10574	876	9945
40	853	9688	1038	11785	1134	12871	1245	14133	1285	14600	1296	14713	1291	14662	1248	14177	1165	13235	1050	11929	881	10002	816	9266
45	794	9022	915	10399	1095	12437	1215	13804	1269	14410	1295	14710	1255	14260	1224	13907	1124	12767	1018	11490	822	9341	790	8971
50	674	7657	862	9795	1052	11951	1187	13482	1259	14297	1275	14483	1257	14271	1204	13676	1086	12338	906	10290	780	8852	642	7301
55	630	7152	809	9189	1006	11420	1147	13029	1230	13969	1267	14384	1239	14076	1175	13345	1047	11889	854	9700	651	7396	618	7014
60	489	5554	765	8690	957	10874	1116	12671	1211	13751	1258	14285	1234	14019	1149	13050	1007	11436	802	9113	621	7062	480	5447

Table 3-13
Monthly Average Daily Radiation on a Tilted Surface
$\overline{H}_{TILT}$ for $\overline{K}_T$ = .4, Tilt = Latitude + 15°, A = Btu/ft^2, B = kJ/m^2

LAT	JAN		FEB		MAR		APRIL		MAY		JUNE		JULY		AUG		SEPT		OCT		NOV		DEC	
	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B
20	1173	13322	1219	13843	1245	14134	1215	13805	1178	13376	1144	12995	1151	13068	1200	13631	1219	13838	1217	13817	1182	13425	1160	13168
25	1136	12907	1178	13382	1215	13796	1215	13805	1178	13381	1143	12975	1158	13149	1186	13468	1206	13695	1181	13412	1141	12953	1038	11788
30	1008	11442	1130	12839	1173	13328	1193	13554	1171	13295	1147	13025	1157	13143	1177	13362	1171	13298	1141	12953	1015	11523	979	11119
35	948	10769	1089	12366	1142	12965	1175	13346	1156	13123	1144	12997	1149	13054	1159	13163	1137	12915	1093	12410	965	10960	923	10479
40	890	10109	1059	12028	1095	12438	1148	13035	1147	13030	1135	12894	1135	12890	1134	12875	1103	12526	1050	11928	909	10329	863	9811
45	837	9502	928	10538	1060	12042	1111	12621	1132	12860	1121	12727	1129	12818	1113	12642	1066	12107	1012	11490	855	9714	846	9606
50	709	8052	878	9969	1013	11508	1088	12358	1117	12624	1102	12509	1103	12527	1085	12322	1033	11731	900	10223	816	9266	686	7788
55	666	7566	821	9326	971	11032	1044	11855	1086	12334	1094	12422	1088	12355	1061	12049	990	11249	854	9700	680	7721	661	7512
60	516	5866	780	8859	928	10542	1009	11469	1058	12014	1072	12175	1058	12016	1030	11696	950	10787	802	9113	652	7413	515	5847

Table 3-14
Monthly Average Daily Radiation on a Tilted Surface
$\overline{H}_{TILT}$ for $\overline{K}_T$ = .4, Vertical, A = Btu/ft^2, B = kJ/m^2

LAT	JAN		FEB		MAR		APRIL		MAY		JUNE		JULY		AUG		SEPT		OCT		NOV		DEC	
	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B
20	892	10125	813	9229	688	7822	530	6022	466	5288	455	5167	452	5134	493	5604	622	7060	763	8667	862	9794	980	10309
25	918	10422	832	9447	728	8278	601	6828	499	5667	480	5446	488	5545	546	6204	675	7669	798	9059	886	10063	830	9430
30	822	9339	850	9655	768	8719	642	7287	557	6331	516	5861	536	6091	608	6906	707	8032	815	9252	812	9219	819	9305
35	814	9251	869	9874	792	8994	700	7951	613	6957	579	6580	582	6609	651	7395	751	8527	830	9432	809	9185	811	9211
40	800	9086	894	10159	819	9301	737	8370	664	7535	640	7273	653	7411	701	7956	780	8862	847	9614	800	9087	792	8993
45	778	8830	799	9072	834	9476	776	8809	723	8212	699	7934	705	8011	742	8428	804	9135	860	9771	780	8860	809	9183
50	674	7657	786	8926	842	9560	813	9238	777	8822	754	8559	754	8563	787	8937	828	9405	788	8953	772	8770	661	7513

Table 3-15

Monthly Average Daily Radiation on a Tilted Surface

$\overline{H}_{TILT}$ for $\overline{K}_T = .5$, Tilt = Latitude - 15°, A = Btu/ft^2, B = kJ/m^2

LAT	JAN		FEB		MAR		APRIL		MAY		JUNE		JULY		AUG		SEPT		OCT		NOV		DEC	
	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B
20	1267	14387	1417	16090	1564	17768	1668	18945	1729	19636	1723	19572	1712	19446	1684	19121	1585	18003	1452	16486	1308	14856	1224	13908
25	1231	13983	1374	15603	1532	17409	1683	19112	1733	19678	1728	19623	1726	19605	1682	19105	1568	17804	1411	16030	1261	14323	1128	12808
30	1120	12725	1334	15151	1508	17128	1668	18947	1725	19587	1738	19742	1729	19635	1686	19142	1551	17611	1377	15647	1161	13186	1076	12222
35	1066	12106	1290	14651	1466	16644	1641	18636	1723	19567	1756	19948	1739	19743	1679	19043	1518	17241	1333	15140	1194	12542	1014	11514
40	1007	11433	1252	14223	1429	16224	1631	18523	1728	19624	1765	20045	1738	19737	1656	18806	1483	16839	1293	14688	1051	11931	960	10901
45	951	10798	1129	12824	1391	15793	1606	18242	1722	19562	1764	20041	1728	19627	1639	18613	1454	16509	1247	14167	993	11275	927	10526
50	815	9257	1072	12172	1345	15270	1566	17789	1701	19323	1757	19956	1728	19624	1609	18279	1414	16054	1132	12862	947	10755	770	8745
55	769	8728	1016	11543	1300	14761	1537	17460	1685	19133	1745	19818	1722	19550	1597	18139	1369	15547	1078	12262	801	9097	740	8408
60	599	6806	971	11032	1255	14255	1503	17068	1673	18999	1751	19886	1713	19449	1572	17852	1330	15106	1026	11661	769	8736	579	6580

Table 3-16

Monthly Average Daily Radiation on a Tilted Surface

$\overline{H}_{TILT}$ for $\overline{K}_T = .5$, Tilt = Latitude, A = Btu/ft^2, B = kJ/m^2

LAT	JAN		FEB		MAR		APRIL		MAY		JUNE		JULY		AUG		SEPT		OCT		NOV		DEC	
	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B
20	1413	16253	1533	17407	1601	18183	1635	18570	1609	18275	1622	18421	1593	18085	1617	18364	1601	18179	1548	17585	1454	16507	1404	15949
25	1400	15895	1485	16869	1576	17902	1634	18555	1629	18497	1604	18220	1604	18219	1615	18348	1583	17975	1502	17059	1415	16066	1278	14508
30	1259	14303	1447	16434	1549	17594	1620	18400	1620	18399	1595	18114	1605	18232	1602	18203	1565	17775	1462	16599	1278	14515	1215	13800
35	1201	13642	1402	15920	1517	17228	1594	18103	1619	18381	1612	18302	1597	18131	1595	18118	1532	17398	1421	16133	1232	13989	1161	13183
40	1139	12937	1369	15544	1476	16765	1571	17837	1606	18249	1601	18184	1596	18126	1576	17902	1496	16987	1382	15690	1173	13320	1098	12463
45	1083	12298	1221	13871	1434	16287	1548	17585	1586	18033	1601	18181	1587	18205	1562	17735	1454	16509	1342	15242	1110	12610	1076	12219
50	926	10514	1167	13258	1384	15713	1511	17164	1574	17871	1594	18104	1588	18038	1535	17433	1424	16181	1206	13698	1070	12151	887	10076
55	878	9974	1107	12577	1343	15247	1486	16873	1554	17647	1583	17979	1567	17790	1512	17167	1379	15661	1158	13158	902	10243	863	9797
60	686	7791	1065	12092	1292	14670	1455	16521	1545	17551	1572	17857	1560	17716	1477	16774	1339	15208	1103	12526	873	9909	676	7673

Table 3-17
Monthly Average Daily Radiation on a Tilted Surface
$\overline{H}_{TILT}$ for $\overline{K}_T$ = .5, Tilt = Latitude + 15°, A = Btu/ft^2, B = kJ/m^2

LAT	JAN		FEB		MAR		APRIL		MAY		JUNE		JULY		AUG		SEPT		OCT		NOV		DEC	
	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B
20	1527	17319	1566	17760	1672	17840	1521	17257	1457	16525	1397	15853	1406	15946	1485	16849	1540	17474	1550	17585	1528	17333	1507	17098
25	1496	16970	1524	17290	1780	20201	1521	17257	1457	16529	1394	15819	1414	16040	1484	16835	1524	17290	1517	17207	1493	1693[illegible]	1359	15415
30	1326	15039	1482	16820	1509	17128	1493	16943	1448	16421	1399	15875	1413	16028	1472	16703	1494	16952	1475	16736	1338	15180	1303	14786
35	2532	28730	1444	16381	1480	16790	1486	16861	1429	16207	1396	15835	1403	15915	1450	16454	1465	16614	1433	16257	1284	14568	1243	14100
40	1207	13689	1415	16052	1430	16224	1451	16465	1418	16091	1402	15911	1403	15911	1437	16275	1432	16248	1392	15801	1232	13974	1177	13353
45	1152	13077	1246	14132	1392	15793	1420	16105	1417	16075	1384	15702	1395	15821	1409	15979	1395	15821	1352	15340	1170	13277	1166	13224
50	982	11144	1194	13548	1346	15270	1389	15760	1391	15780	1378	15635	1380	15659	1388	15740	1348	15296	1215	13782	1130	12824	952	10799
55	935	10613	1139	12921	1301	14761	1358	15406	1375	15603	1350	15324	1361	15444	1356	15386	1310	14861	1166	13227	949	10761	933	10588
60	730	8284	1096	12432	1264	14338	1324	15019	1340	15200	1342	15219	1341	15212	1329	15081	1278	14498	1114	12633	924	10483	729	8226

Table 3-18
Monthly Average Daily Radiation on a Tilted Surface
$\overline{H}_{TILT}$ for $\overline{K}_T$ = .5, Vertical, A = Btu/ft^2, B = kJ/m^2

LAT	JAN		FEB		MAR		APRIL		MAY		JUNE		JULY		AUG		SEPT		OCT		NOV		DEC	
	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B
20	1174	13322	1044	11840	862	9777	612	6940	496	5638	500	5676	497	5634	550	6247	762	8649	983	11148	1140	12931	1215	13780
25	1232	13982	1090	12370	927	10511	703	7979	555	6297	512	5807	541	6139	634	7188	830	9415	1024	11618	1175	13326	1109	12581
30	1103	12515	1132	12839	975	11055	787	8926	645	7320	574	6513	600	6812	711	8070	899	10204	1068	12118	1084	12300	1112	12617
35	1107	12558	1169	13266	1030	11680	860	9762	714	8104	652	7403	675	7655	782	8874	953	10815	1105	12534	1089	12349	1110	12600
40	1100	12486	1209	13715	1072	12168	922	10462	796	9207	747	8472	763	8660	861	9765	1003	11374	1138	12908	1088	12340	1099	12468
45	1084	12298	1092	12388	1099	12462	985	11175	888	10072	819	9297	847	9613	929	10535	1043	11831	1163	13190	1070	12143	1128	12800

Table 3-19

Monthly Average Daily Radiation on a Tilted Surface

$\overline{H}_{TILT}$ for $\overline{K}_T = .6$, Tilt = Latitude - 15°, A = Btu/ft², B = kJ/m²

LAT	JAN		FEB		MAR		APRIL		MAY		JUNE		JULY		AUG		SEPT		OCT		NOV		DEC	
	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B
20	1536	17425	1702	19308	1887	21407	2004	22734	2057	23330	2070	23486	2052	23336	2023	22946	1923	21815	1760	19971	1572	17828	1485	16842
25	1504	17066	1680	19062	1859	21088	2022	22934	2082	23614	2076	23547	2074	23526	2021	22926	1901	21570	1727	19589	1541	17486	1379	15642
30	1379	15649	1630	18489	1828	20740	2004	22736	2072	23503	2088	23690	2077	23561	2025	22971	1880	21330	1684	19103	1418	16089	1324	15022
35	1107	12558	1598	18134	1807	20498	1990	22576	2070	23480	2088	23690	2088	23691	2014	22851	1857	21066	1641	18615	1378	15630	1271	14417
40	1273	14441	1569	17800	1759	19955	1977	22434	2076	23548	2098	23804	2087	23685	2008	22784	1828	20738	1613	18294	1323	15004	1211	13735
45	1218	13821	1421	16122	1723	19544	1947	22088	2069	23474	2120	24050	2097	23793	1987	22545	1790	20306	1571	17821	1263	14331	1192	13519
50	1055	11970	1372	15562	1685	19121	1931	21909	2071	23500	2110	23947	2976	23548	1969	22342	1765	20024	1440	16337	1225	13400	1002	11362
55	1007	11422	1318	14954	1644	18646	1909	21657	2063	23405	2118	24027	2089	23695	1953	22155	1717	19479	1384	15707	1048	11893	976	11067
60	792	8483	1270	14409	1604	18200	1877	21300	2048	23233	2125	24107	2078	23570	1937	21976	1683	19100	1336	15160	1019	11560	770	8730

Table 3-20

Monthly Average Daily Radiation on a Tilted Surface

$\overline{H}_{TILT}$ for $\overline{K}_T = .6$, Tilt = Latitude, A = Btu/ft², B = kJ/m²

LAT	JAN		FEB		MAR		APRIL		MAY		JUNE		JULY		AUG		SEPT		OCT		NOV		DEC	
	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B
20	1761	19983	1862	21124	1941	22025	1964	22284	1933	21931	1884	21373	1892	21469	1942	22036	1942	22027	1876	21290	1790	20303	1727	19598
25	1732	19647	1829	20749	1928	21877	1963	22267	1936	21960	1885	21384	1906	21624	1941	22018	1920	21775	1851	21000	1752	19877	1582	17954
30	1568	17795	1793	20338	1894	21487	1946	22080	1925	21842	1894	21493	1907	21638	1926	21845	1898	21528	1814	20573	1606	18216	1532	17387
35	1520	17238	1757	19934	1869	21200	1934	21937	1923	21821	1914	21716	1918	21758	1916	21741	1874	21254	1772	20104	1551	17598	1474	16720
40	1463	16607	1730	19628	1831	20767	1923	21816	1930	21900	1923	21821	1917	21751	1913	[illegible]	1843	20915	1730	19628	1495	16965	1413	16024
45	1402	15901	1560	17692	1788	20284	1895	21496	1926	21847	1923	21818	1907	21629	1895	21492	1819	20636	1705	19345	1424	16253	1399	15868
50	1207	13696	1502	17040	1756	19918	1865	21160	1911	21674	1915	21725	1906	21646	1880	21326	1778	20175	1546	17540	1395	15823	1166	13233
55	1161	13173	1446	16402	1705	19345	1847	20952	1886	21399	1902	21575	1902	21583	1850	20989	1741	19753	1493	16946	1189	13490	1148	13021
60	912	10347	1405	15936	1666	18896	1820	20644	1875	21278	1910	21672	1894	21490	1839	20868	1705	19344	1445	16391	1165	13214	900	10219

Table 3-21
Monthly Average Daily Radiation on a Tilted Surface
$\overline{H}_{TILT}$ for $\overline{K}_T$ = .6, Tilt = Latitude + 15°, A = Btu/ft², B = kJ/m²

LAT	JAN		FEB		MAR		APRIL		MAY		JUNE		JULY		AUG		SEPT		OCT		NOV		DEC	
	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B
20	1902	21582	1927	21858	1905	21613	1825	20708	1707	19364	1636	18559	1666	18901	1762	19992	1867	21180	1910	21667	1891	21459	1890	21435
25	1870	21224	1888	21423	1894	21482	1825	20708	1707	19363	1652	18741	1676	19010	1761	19975	1847	20954	1882	21353	1858	21073	1702	19313
30	1680	19056	1847	20954	1861	21113	1811	20550	1716	19468	1658	18806	1674	18994	1747	19818	1828	20738	1842	20900	1688	19147	1647	18689
35	1625	18430	1818	20626	1822	20674	1784	20233	1714	19449	1653	18754	1662	18856	2112	23960	1791	20313	1798	20402	1633	18524	1589	18022
40	1560	17690	1805	20481	1787	20280	1778	20170	1702	19310	1661	18845	1662	18851	2161	24520	1766	20029	1765	20030	1582	17945	1528	17332
45	1504	17061	1597	18111	1749	19840	1756	19919	1701	19291	1639	18595	1652	18745	1709	19385	1732	19645	1736	19697	1524	17294	1527	17328
50	1288	14612	1548	17562	1709	19387	1717	19474	1669	18936	1632	18516	1656	18791	1683	19092	1698	19266	1563	17741	1488	16878	1259	14282
55	1245	14122	1494	16953	1674	18996	1692	19191	1670	18948	1621	18388	1634	18533	1662	18851	1669	18930	1516	17195	1256	14245	1244	14114
60	973	11040	1454	16497	1631	18499	1661	18842	1646	18673	1631	18506	1629	18486	1644	18652	1630	18493	1462	16585	1235	14015	973	11039

Table 3-22
Monthly Average Daily Radiation on a Tilted Surface
$\overline{H}_{TILT}$ for $\overline{K}_T$ = .6, Vertical, A = Btu/ft², B = kJ/m²

LAT	JAN		FEB		MAR		APRIL		MAY		JUNE		JULY		AUG		SEPT		OCT		NOV		DEC	
	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B
20	1494	16945	1300	14754	1016	11527	674	7653	514	5833	517	5871	514	5834	581	6588	877	9954	781	8855	1426	16177	1552	17608
25	1567	17783	1368	15519	1112	12613	805	9129	604	6848	530	6007	545	6178	700	7945	978	11093	1276	14471	1502	17038	1415	16049
30	1413	16027	1426	16177	1202	13640	906	10274	690	6834	624	7082	636	7213	814	9233	1079	12245	1338	15185	1383	15690	1439	16323
35	1434	16262	1489	16889	1266	14366	1014	11501	794	9013	718	8143	746	8461	919	10427	1160	13166	1404	15934	1408	15977	1447	16420
40	1440	16337	1558	17678	1344	15251	1107	12554	913	10361	830	9423	852	9667	1013	11500	1234	14003	1448	16424	1417	16082	1441	16351
45	1434	16269	1412	16017	1396	15842	1199	13607	1024	11621	939	10661	974	11055	1114	12642	1310	14858	1499	17001	1419	16093	1488	16884

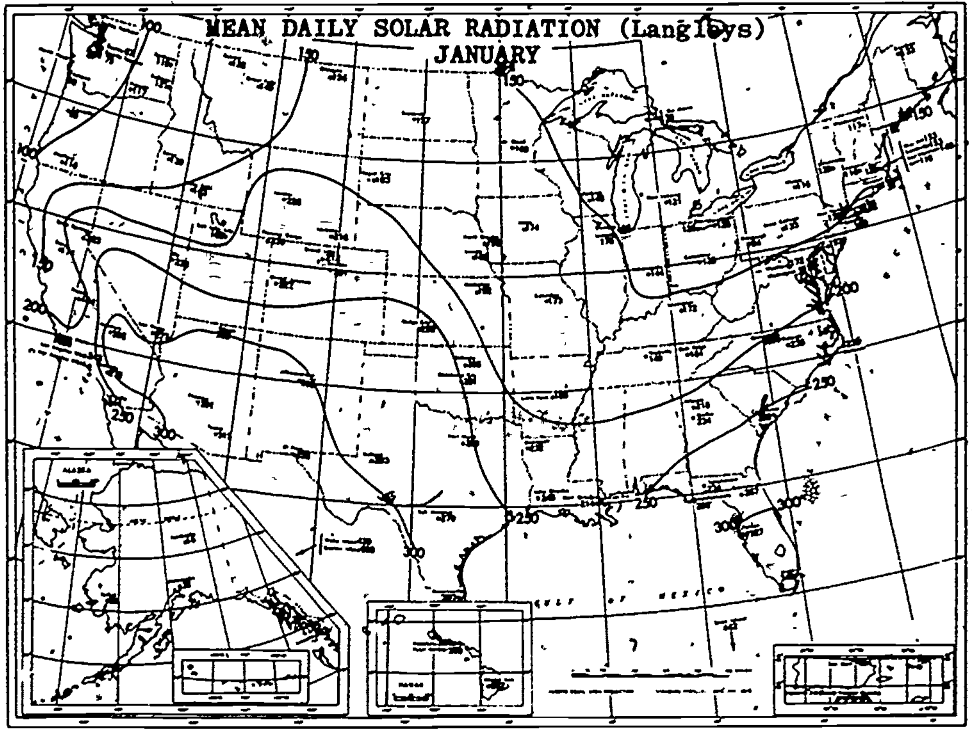

MEAN DAILY SOLAR RADIATION (Langleys)
JANUARY
100
150
200
250
300

Figure 3-1

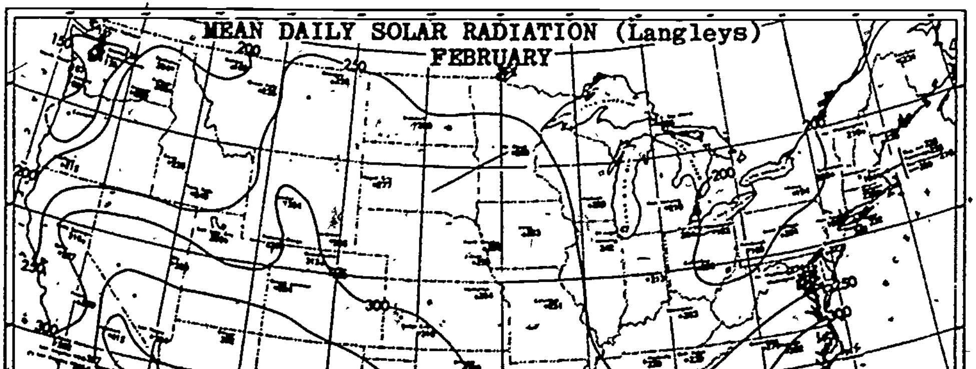

MEAN DAILY SOLAR RADIATION (Langleys)
FEBRUARY
150
200
250
300
350
400

Figure 3-2

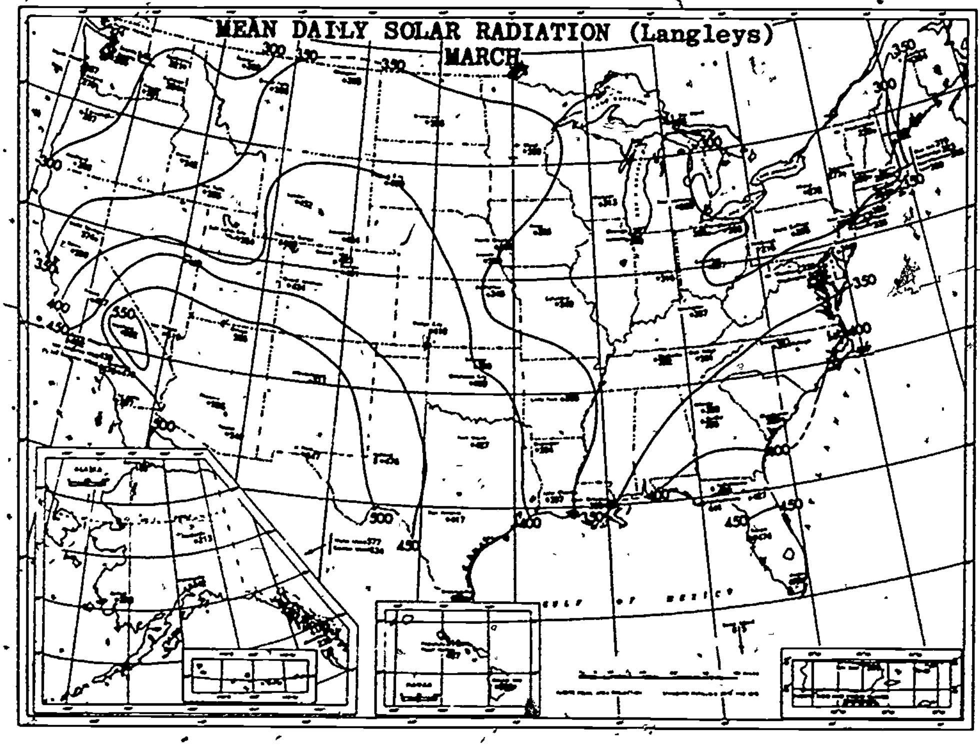

MEAN DAILY SOLAR RADIATION (Langleys)
MARCH
300
350
400
450
500
GULF OF MEXICO

Figure 3-3

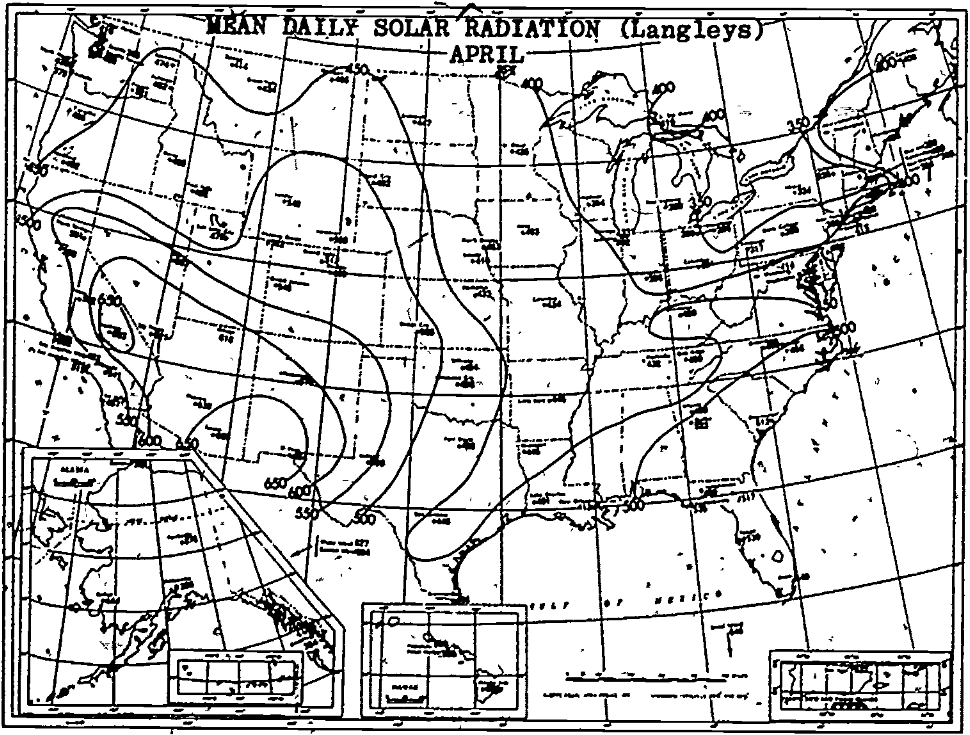

MEAN DAILY SOLAR RADIATION (Langleys)
APRIL
350
400
450
500
550
600
650
GULF OF MEXICO

Figure 3-4

ERIC Full Text Provided by ERIC

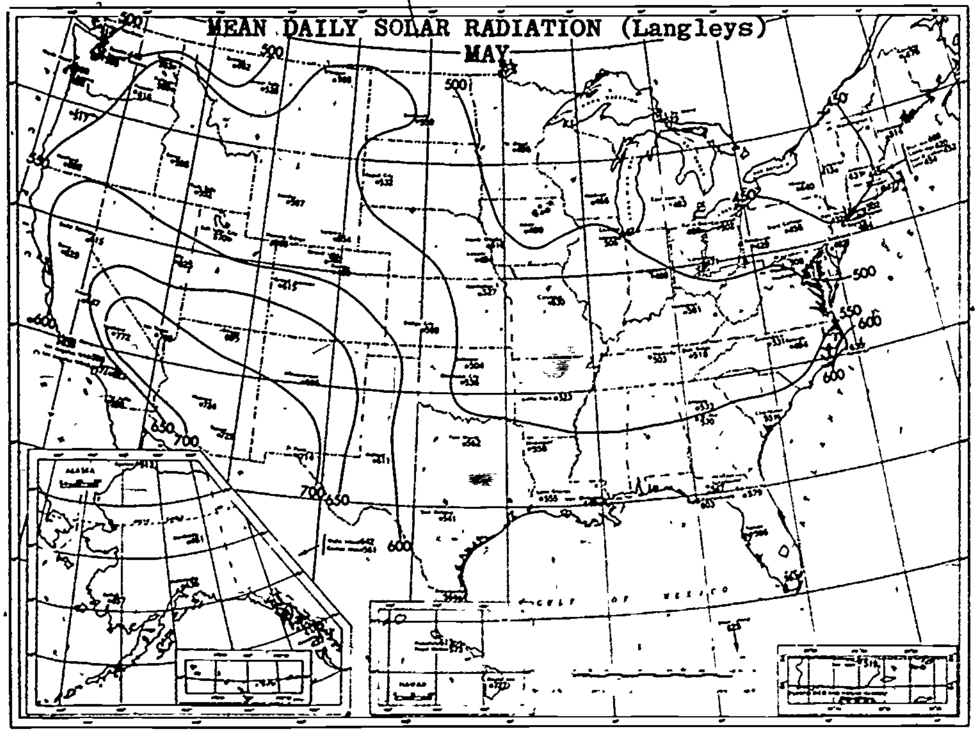

MEAN DAILY SOLAR RADIATION (Langleys)
MAY
500
550
600
650
700

Figure 3-5

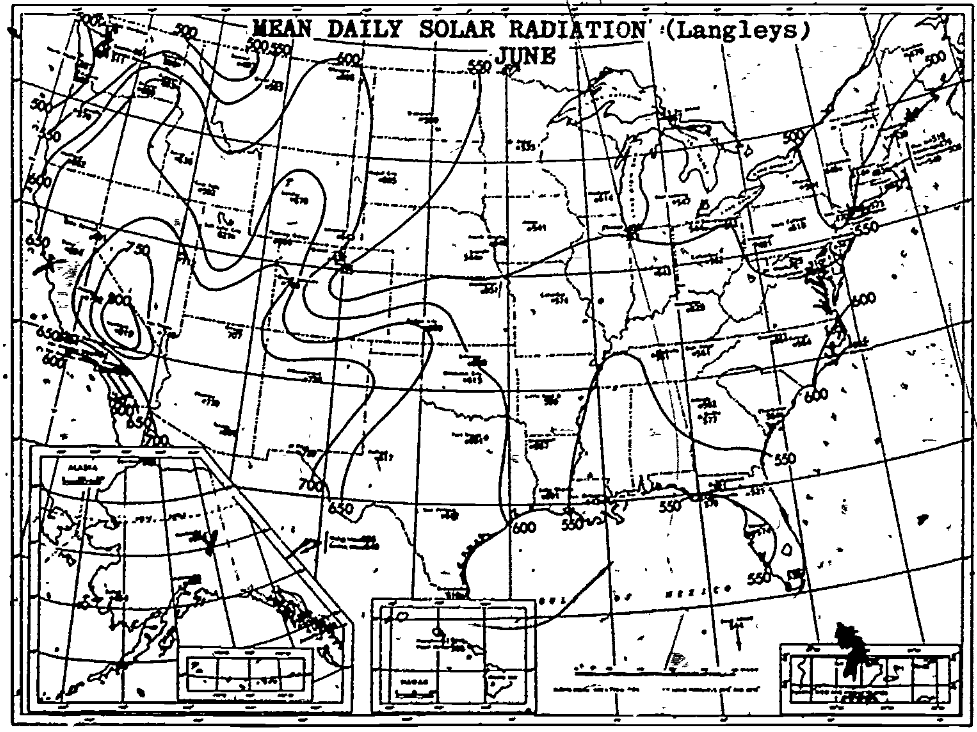

MEAN DAILY SOLAR RADIATION (Langleys)
JUNE
500
550
600
650
700
750
800

Figure 3-6

ERIC
Full Text Provided by ERIC

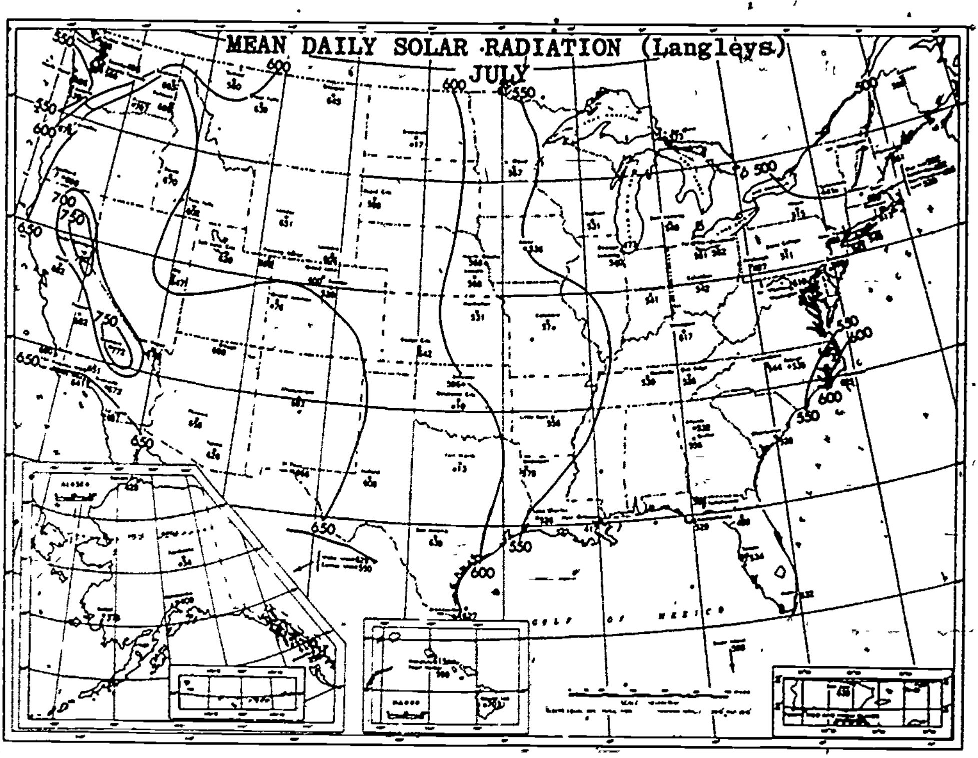

MEAN DAILY SOLAR RADIATION (Langleys)
JULY
550
600
650
700
750
500
GULF OF MEXICO

Figure 3-7

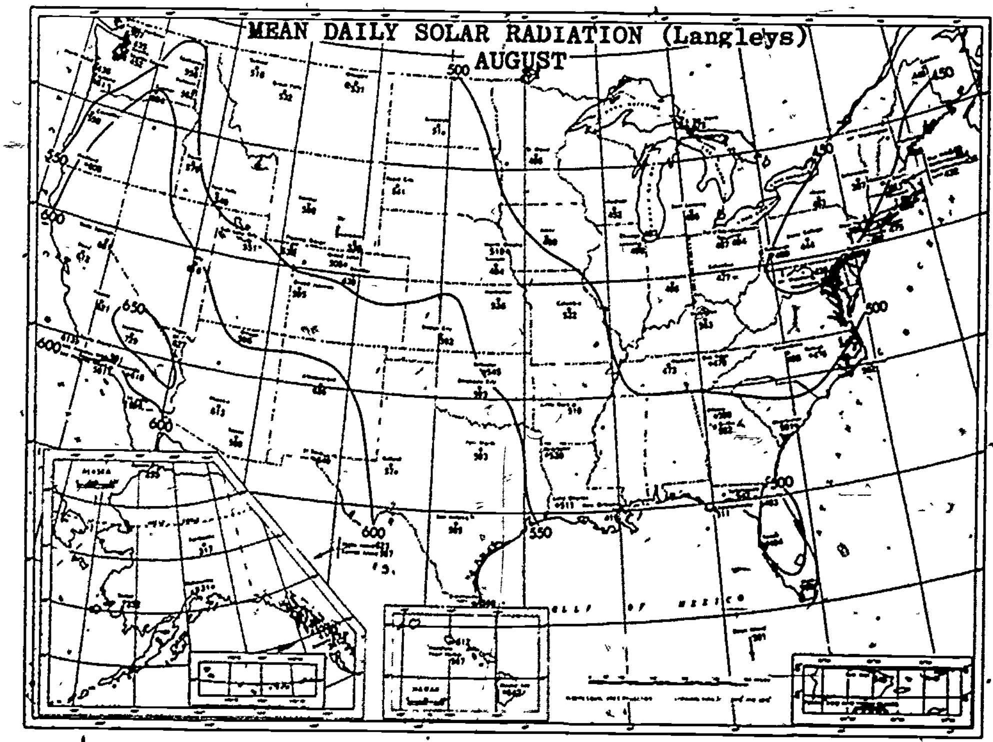

MEAN DAILY SOLAR RADIATION (Langleys)
AUGUST
500
450
550
600
650
GULF OF MEXICO

Figure 3-8

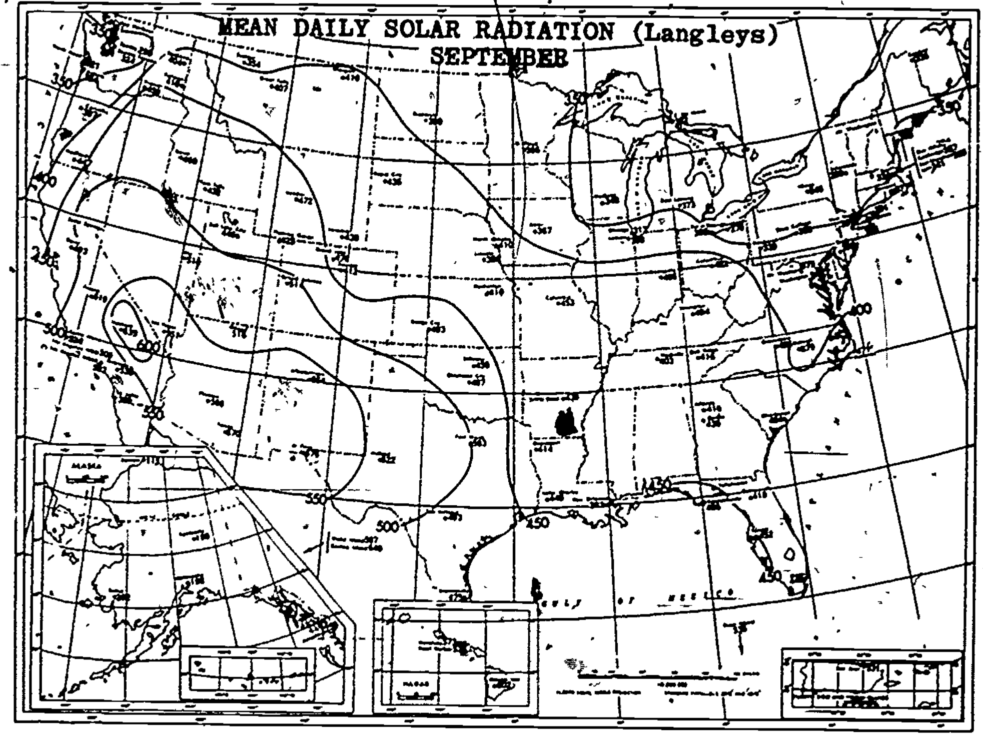

MEAN DAILY SOLAR RADIATION (Langleys)
SEPTEMBER
350
400
450
500
550
600
450
500
550
350
400
450

Figure 3-9

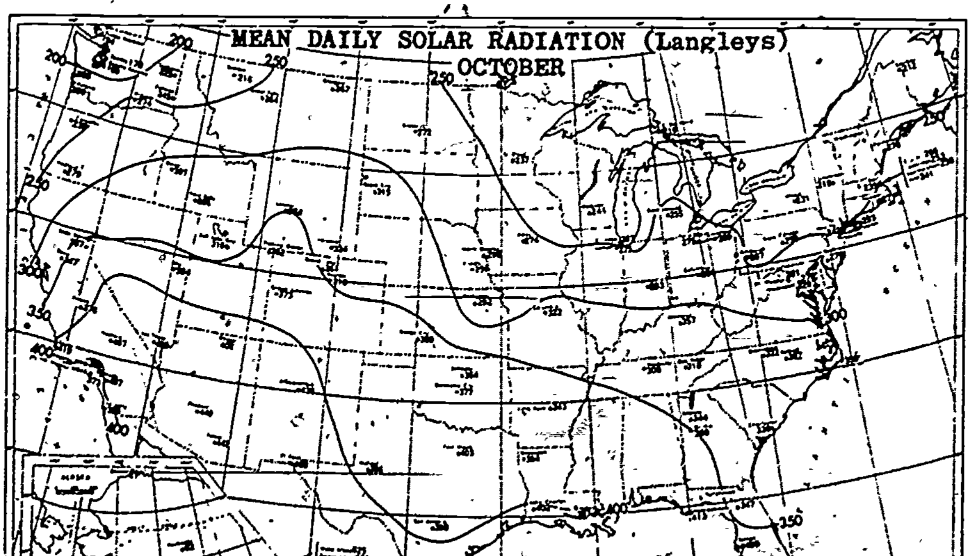
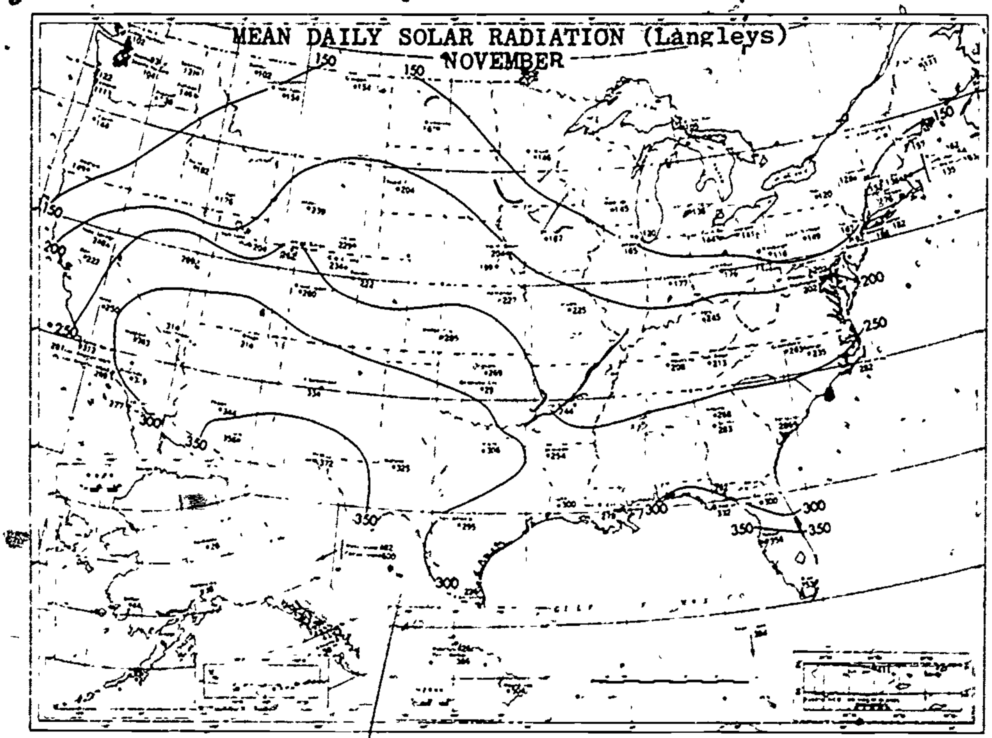

MEAN DAILY SOLAR RADIATION (Langleys)
OCTOBER
200
250
300
350
400
300
350
400
250
300
350

Figure 3-10

ERIC
Full Text Provided by ERIC


MEAN DAILY SOLAR RADIATION (Langleys)
NOVEMBER
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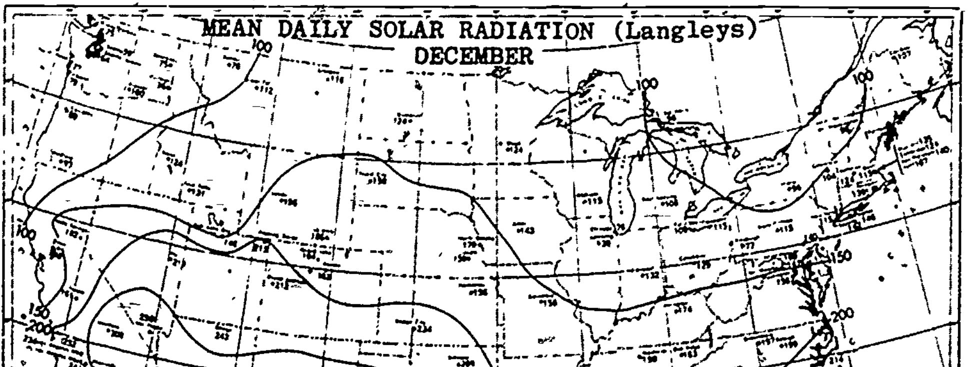
150
150
150
200
200
250
250
300
300
350
350
350
300
350
300

Figure 3-11


MEAN DAILY SOLAR RADIATION (Langleys)
DECEMBER
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100
100
100
150
150
200
200
250
300
300
250
250
250
300
300

Figure 3-12

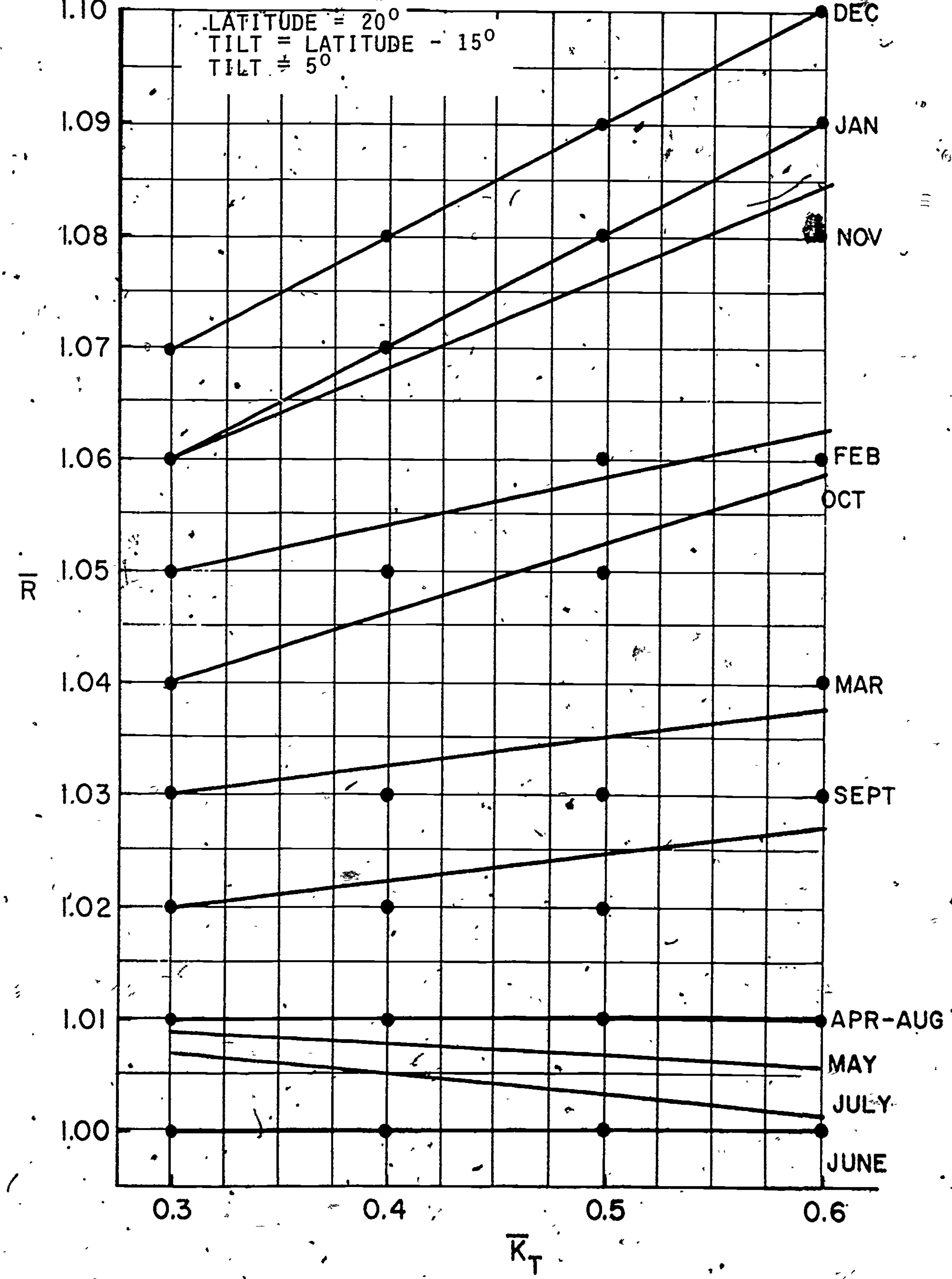
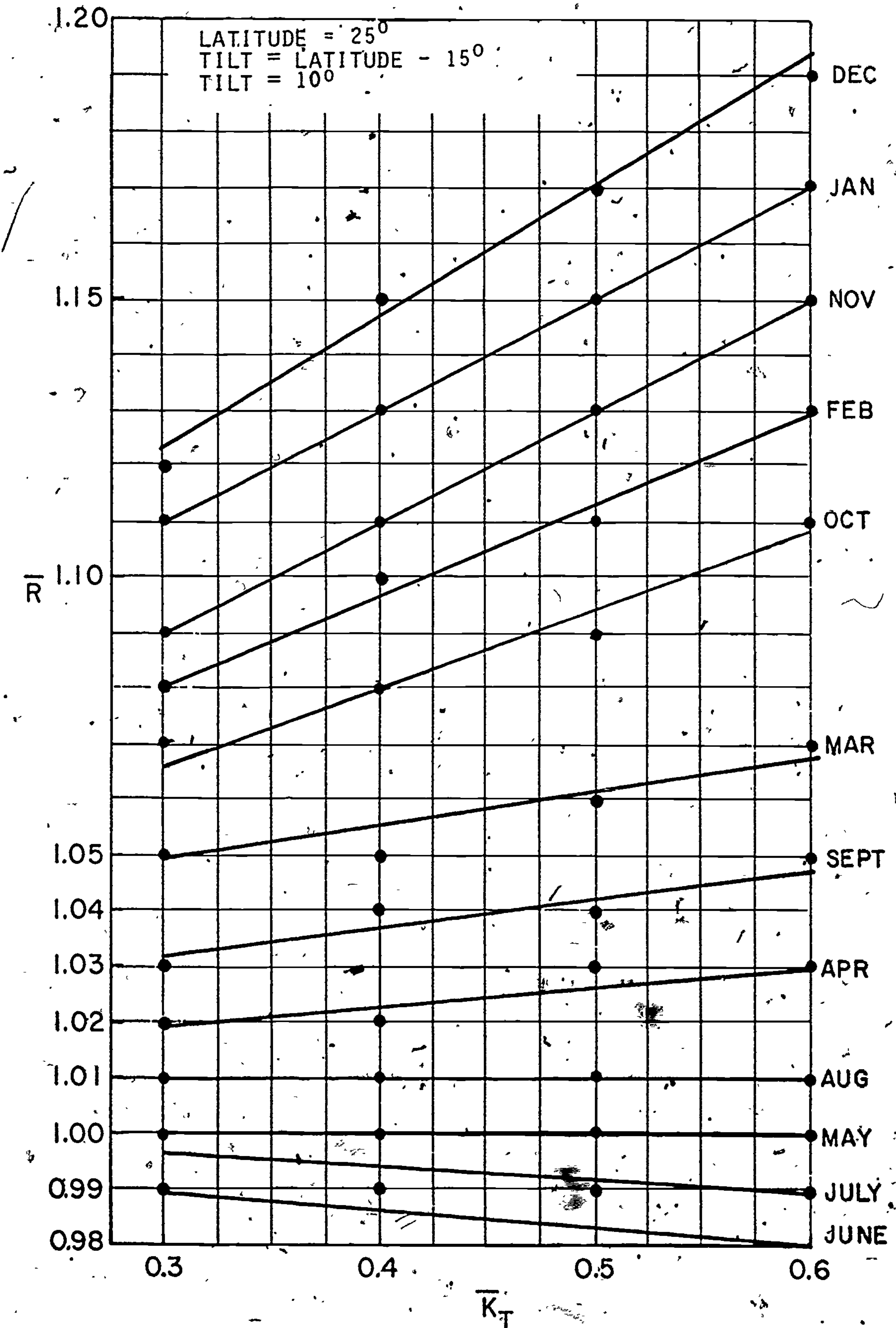


Figure 3-13
$\overline{R}$ versus $\overline{K}_T$



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$\overline{R}$ versus $\overline{K}_T$

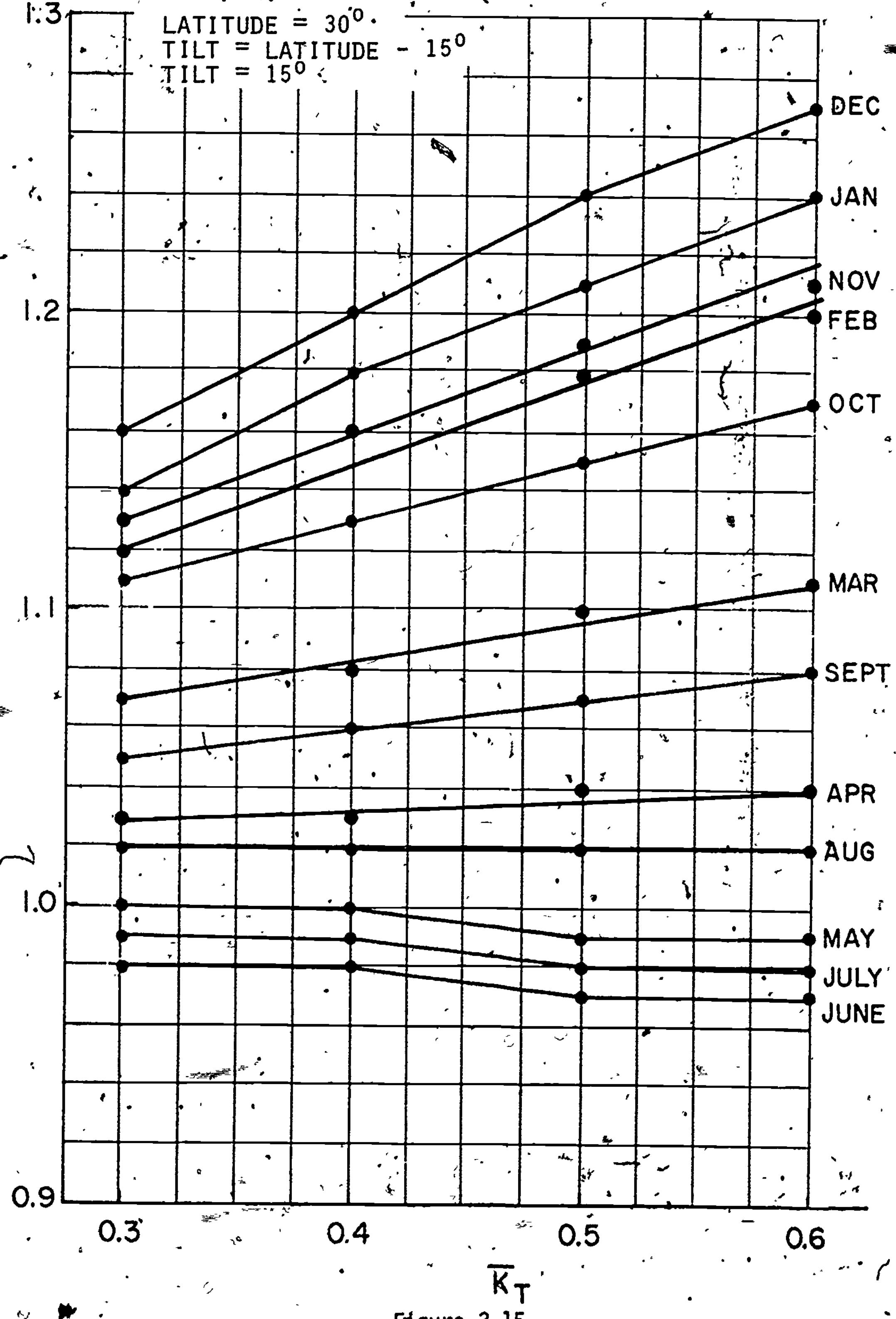
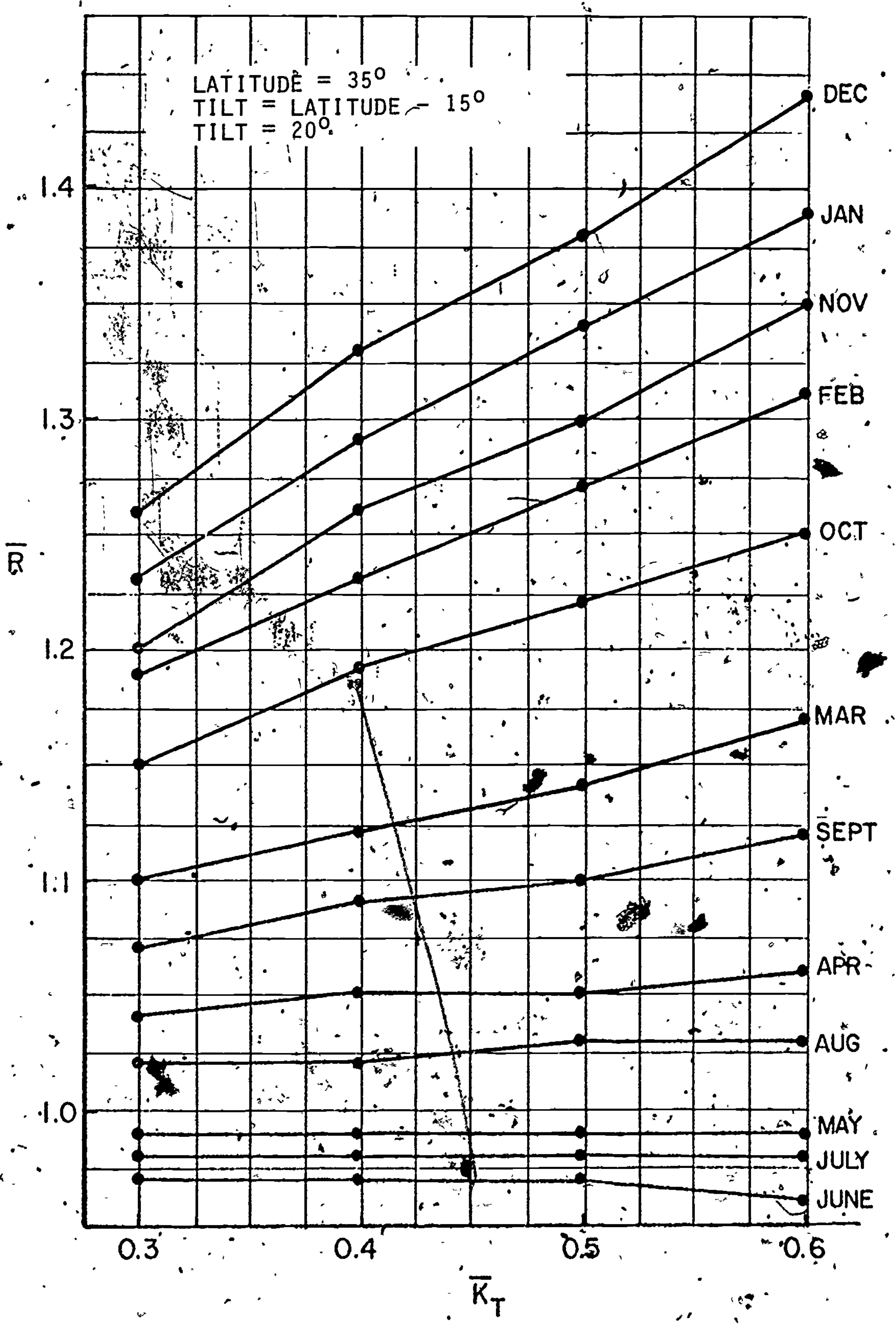


Figure 3-15

$\overline{R}$ versus $\overline{K}_T$



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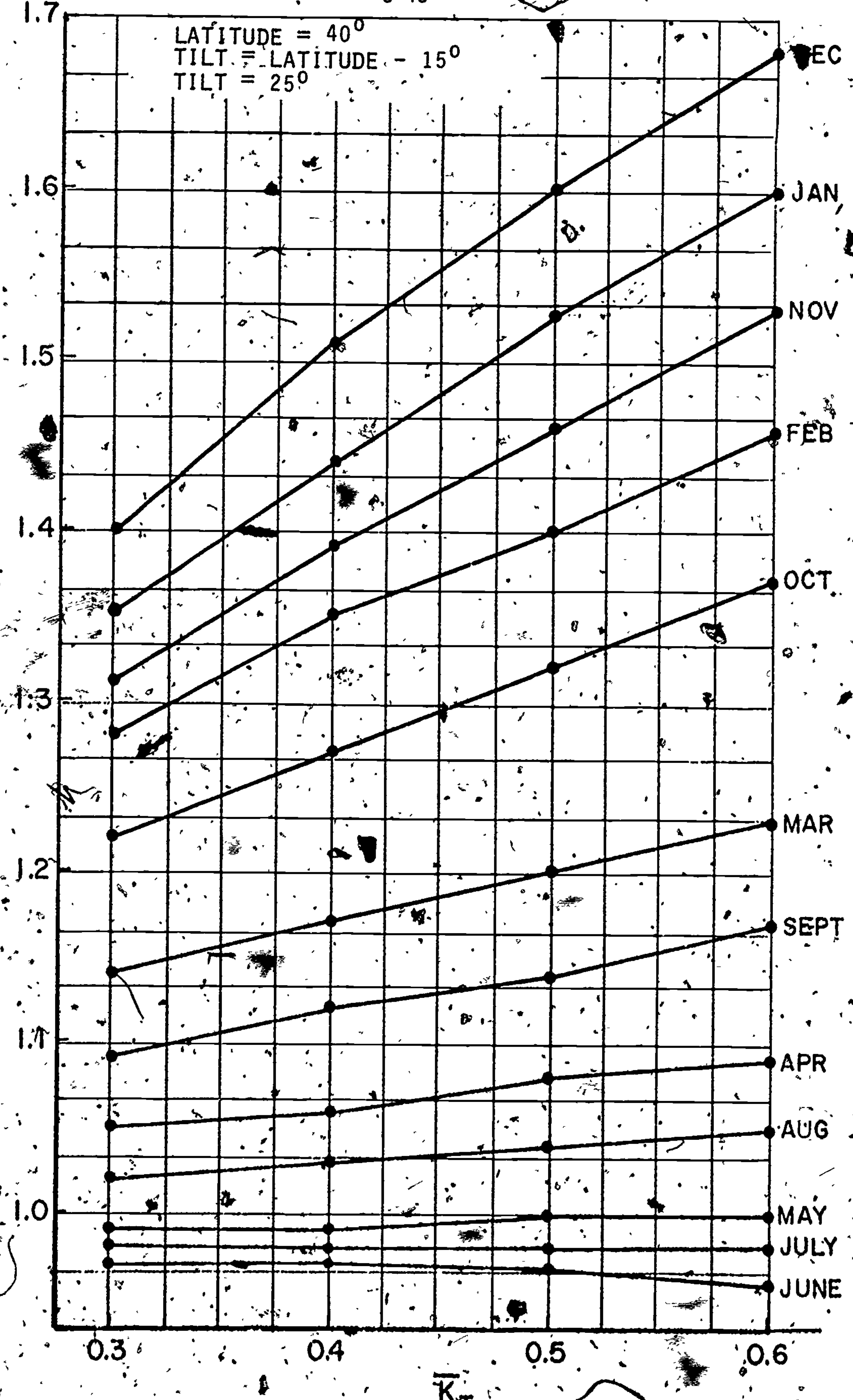
$\overline{R}$ versus $\overline{K}_T$



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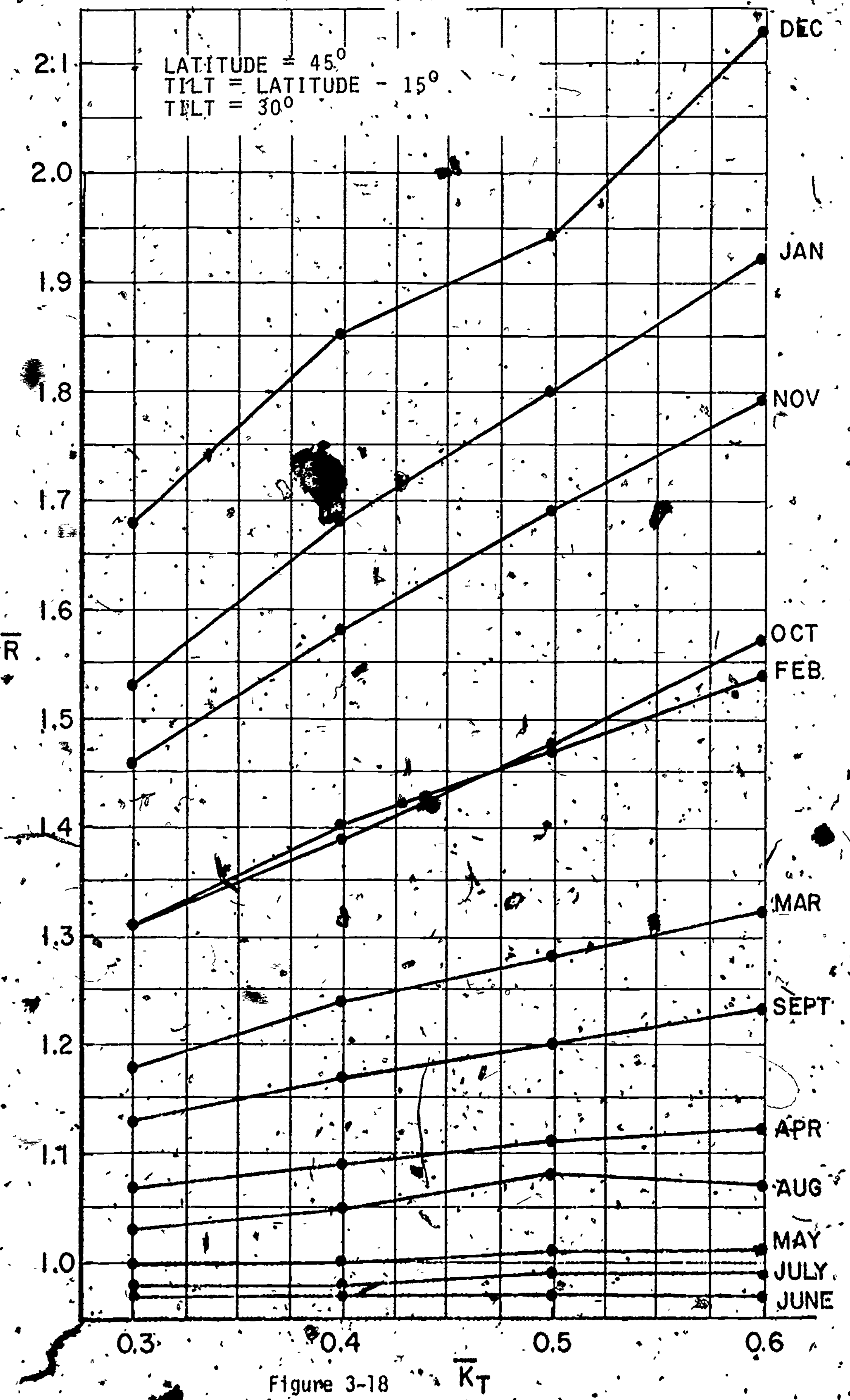
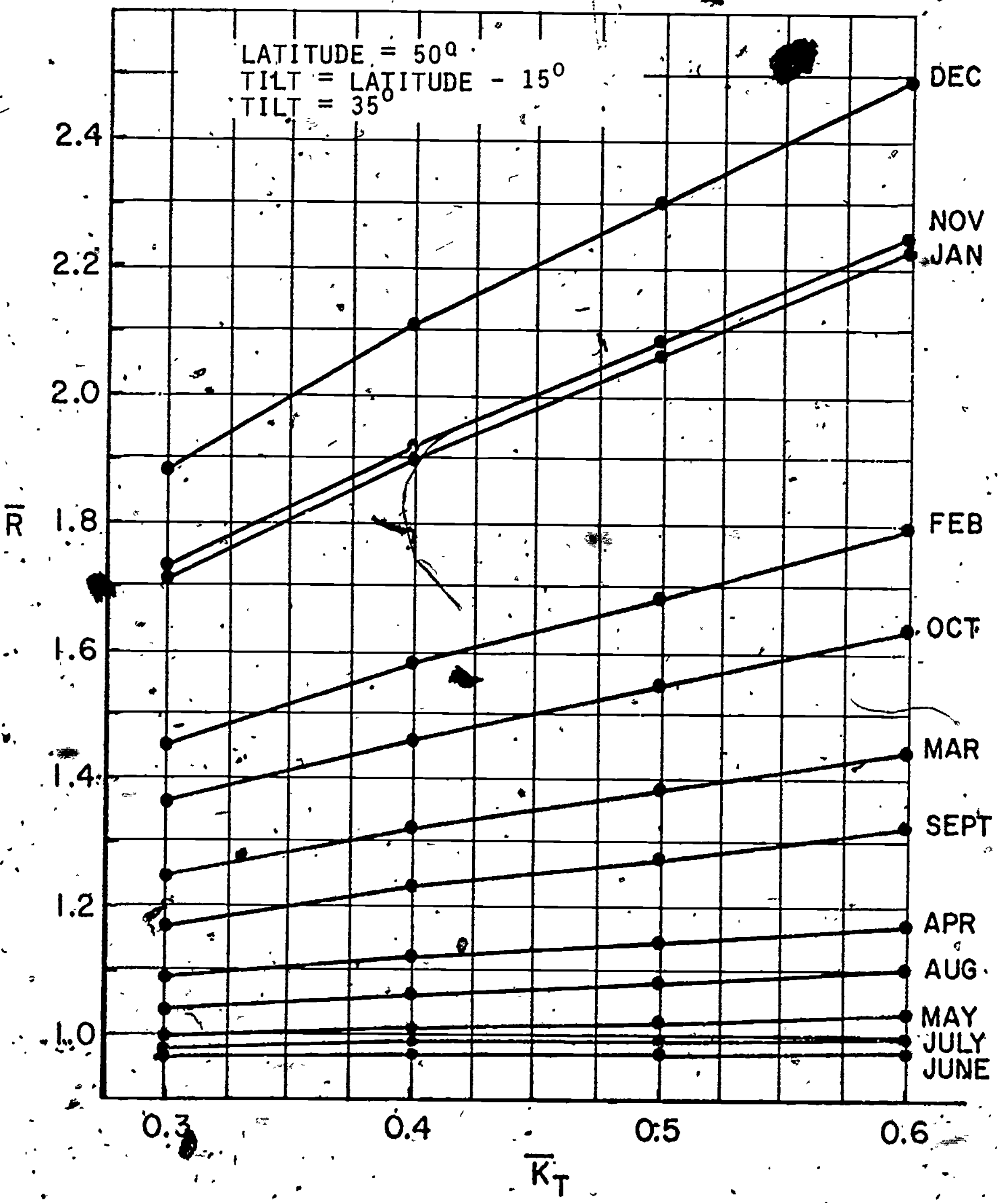


Figure 3-18

$\overline{R}$ versus $\overline{K}_T$



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$\overline{R}$ versus $\overline{K}_T$

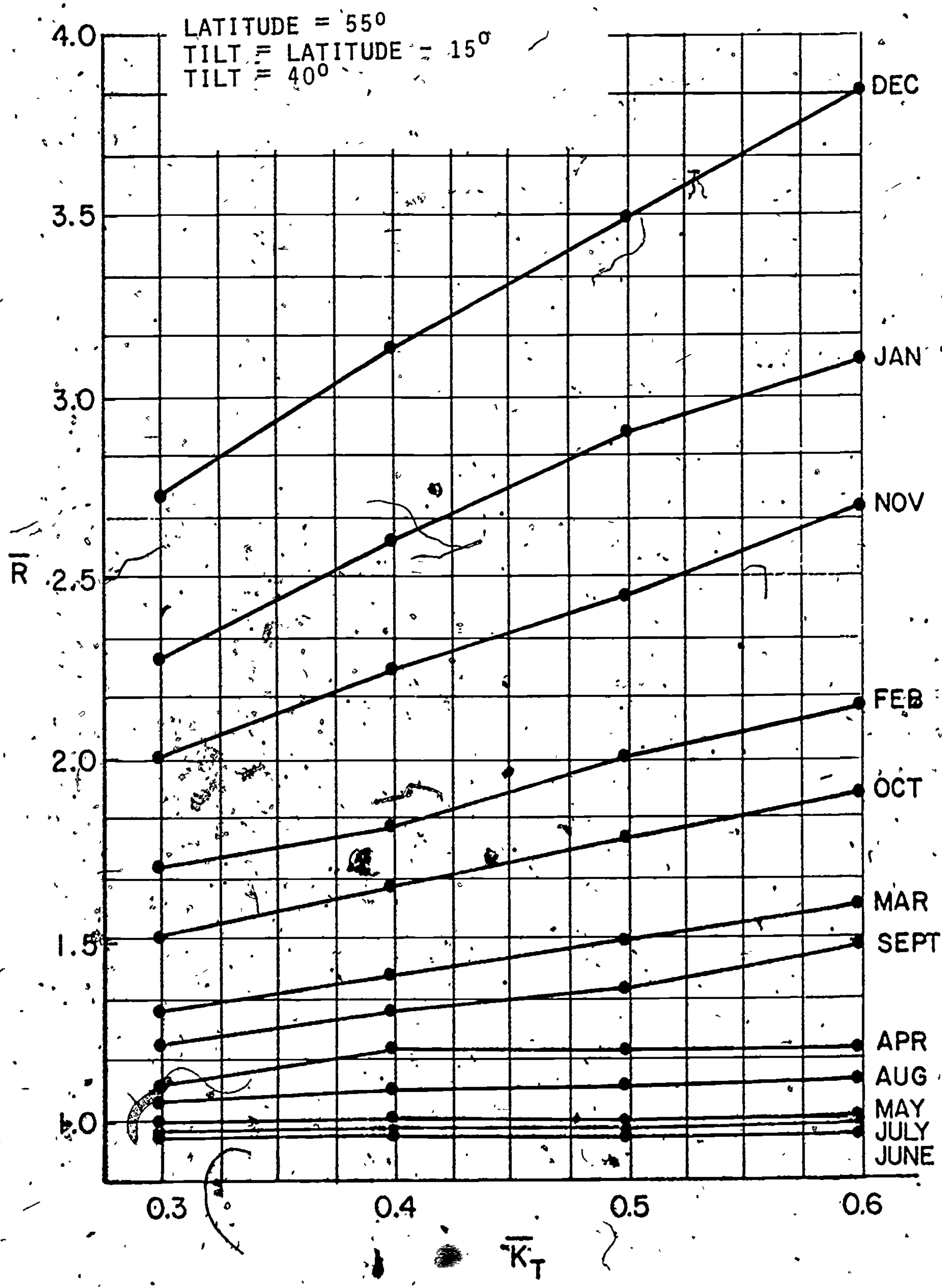


Figure 3-20

$\overline{R}$ versus $\overline{K}_T$

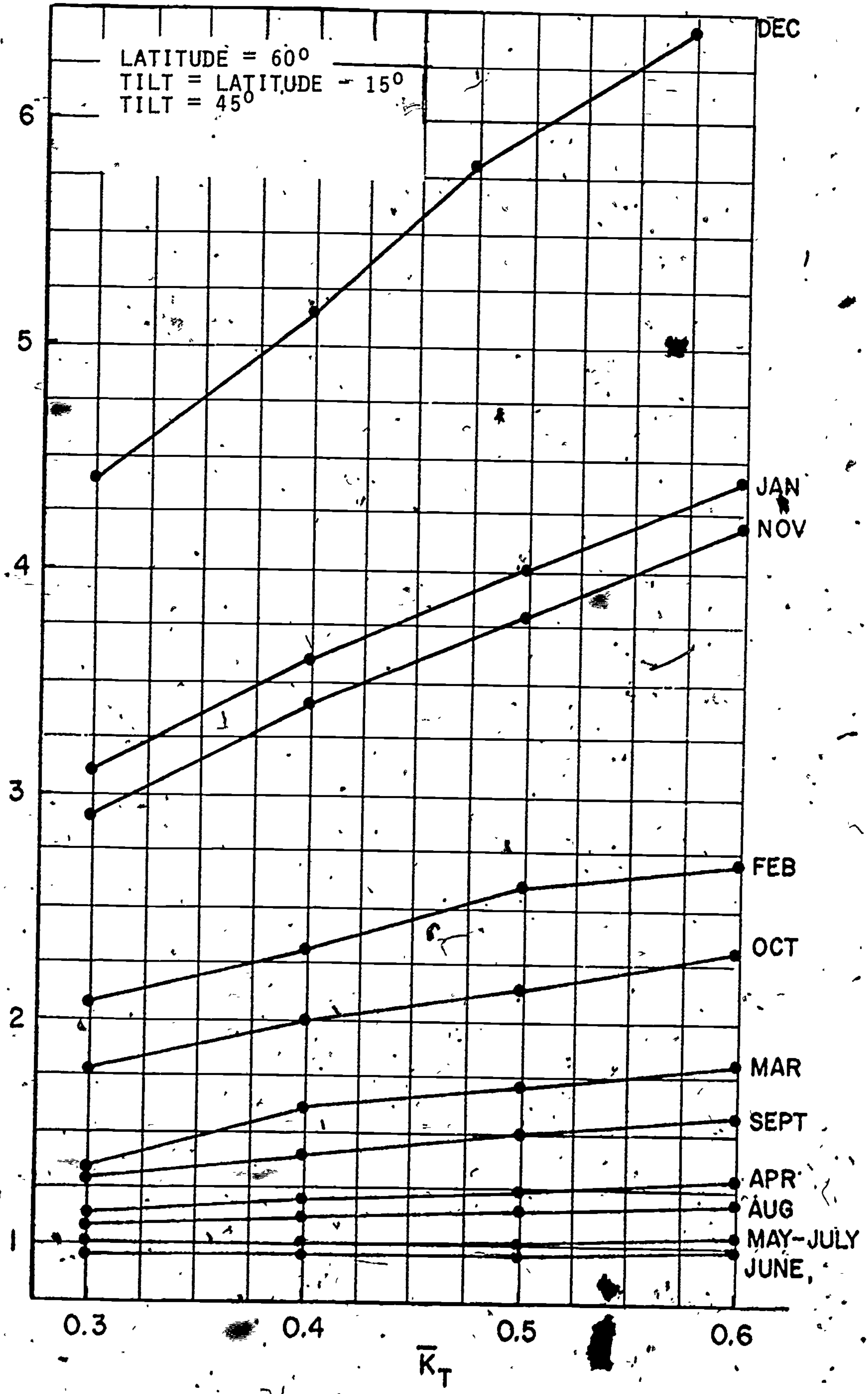


Figure 3-21

$\overline{R}$ versus $\overline{K}_T$.

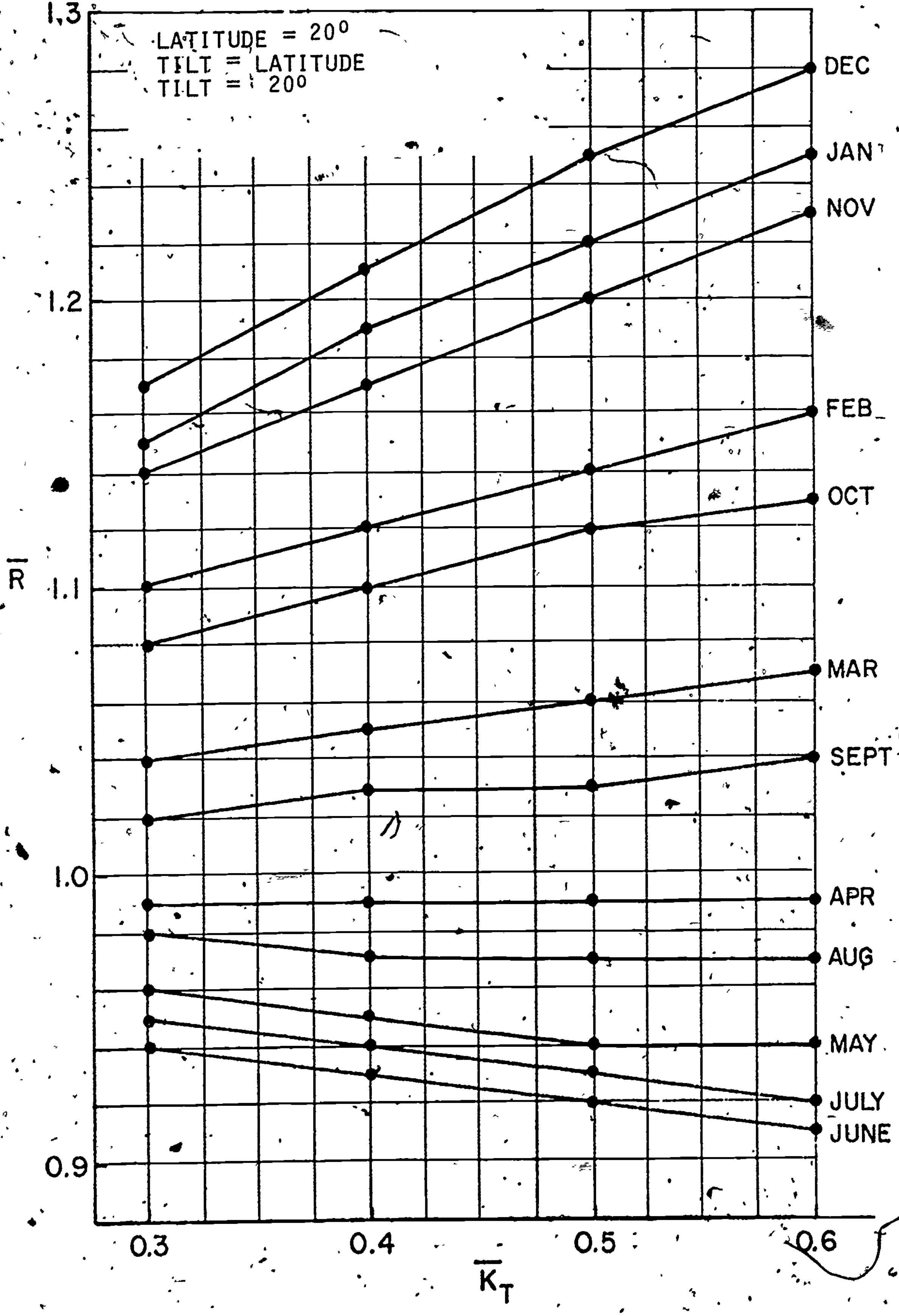
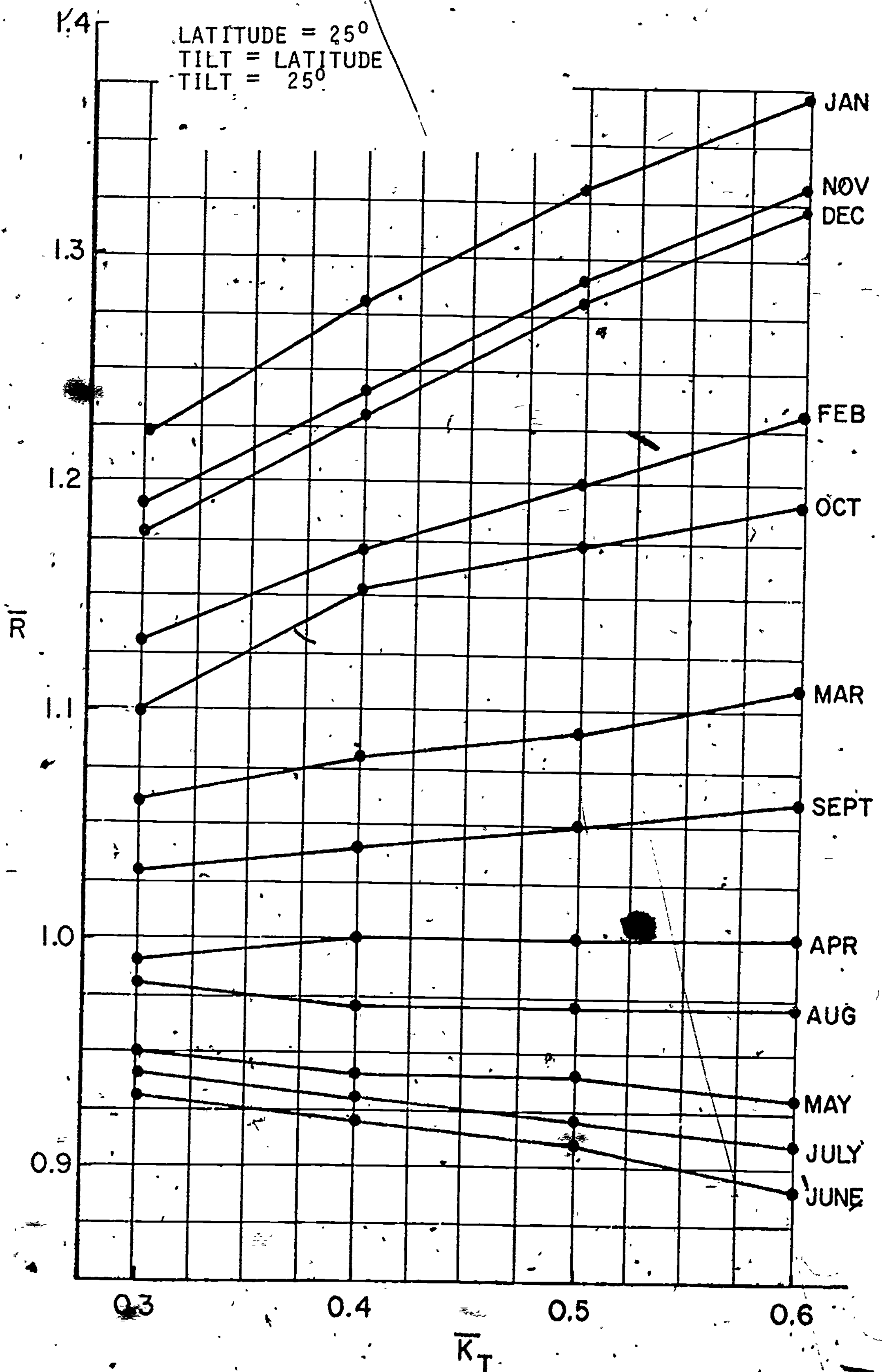


Figure 3-22

$\overline{R}$ versus $\overline{K}_T$



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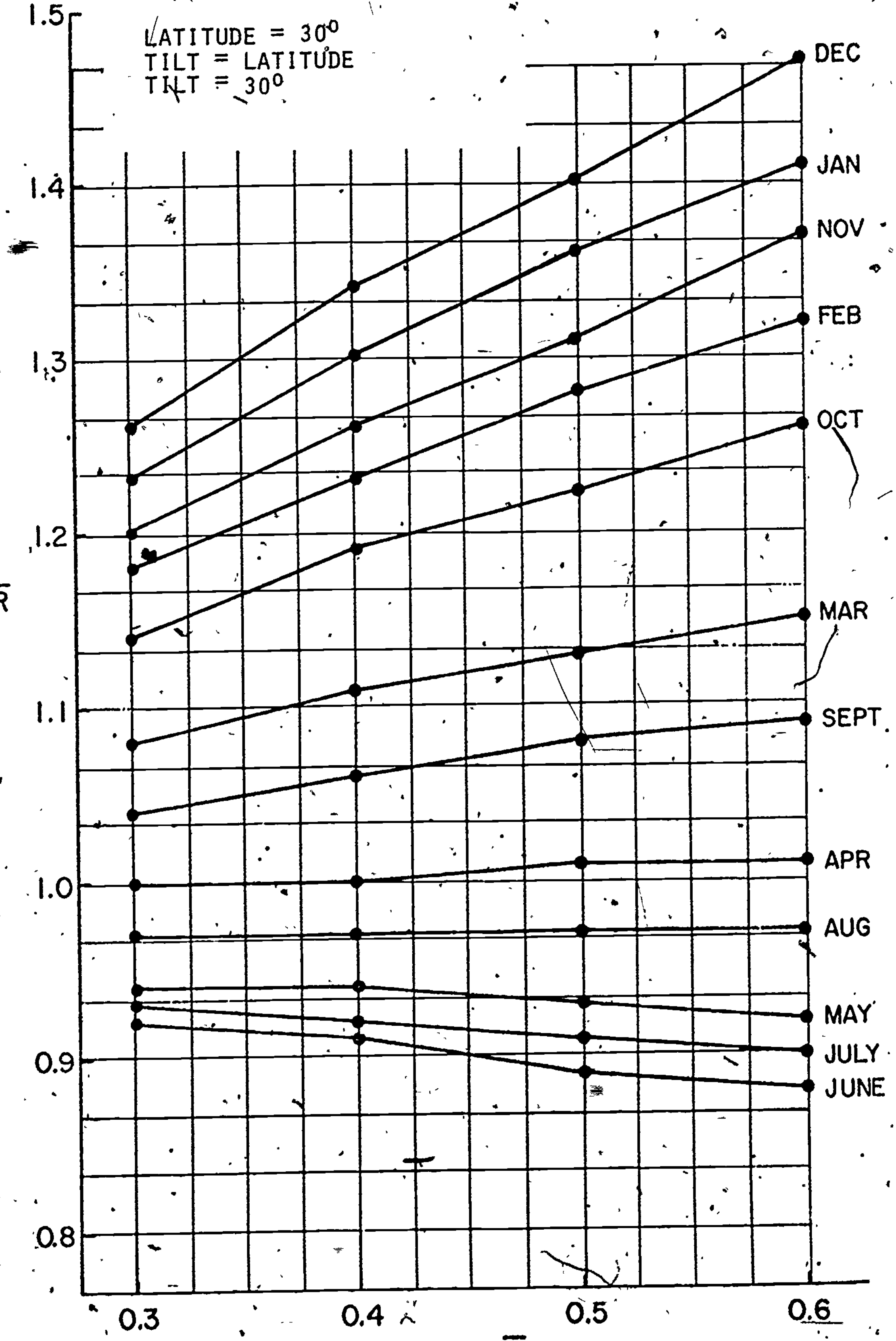


Figure 3-24

$\overline{R}$ versus $\overline{K}_T$

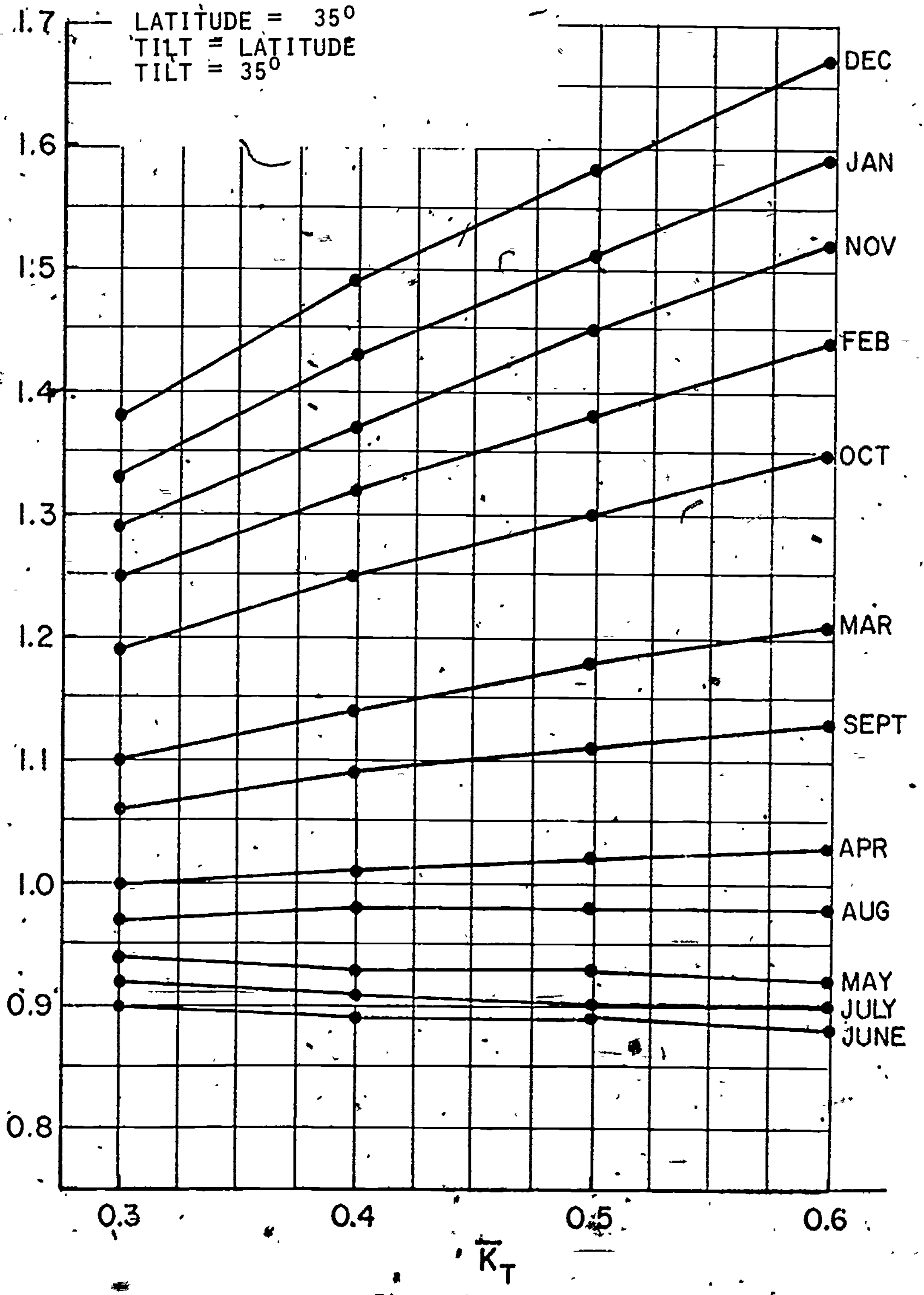


Figure 3-25

$\overline{R}$ versus $\overline{K}_T$

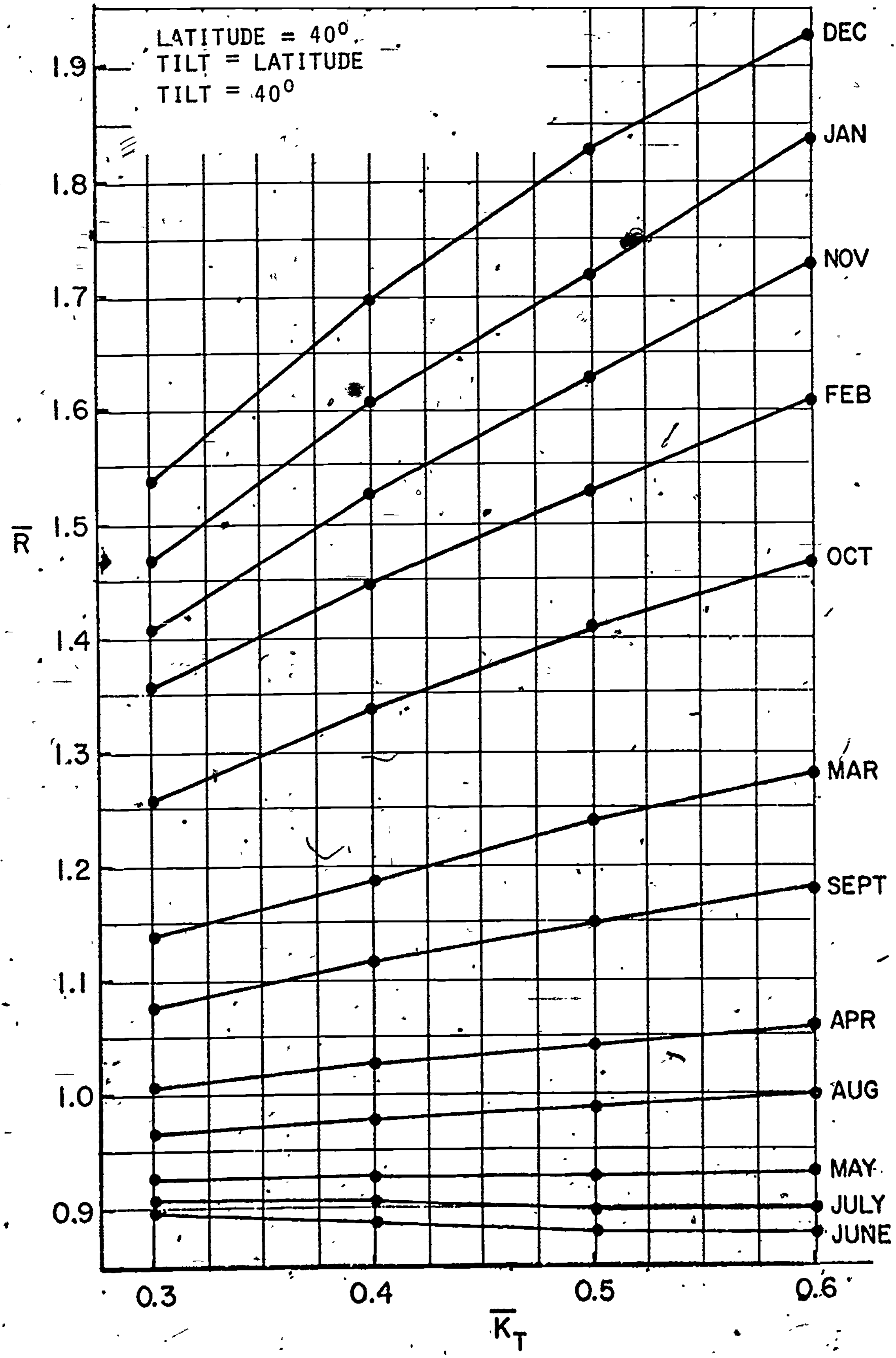


Figure 3-26

$\overline{R}$ versus $\overline{K}_T$

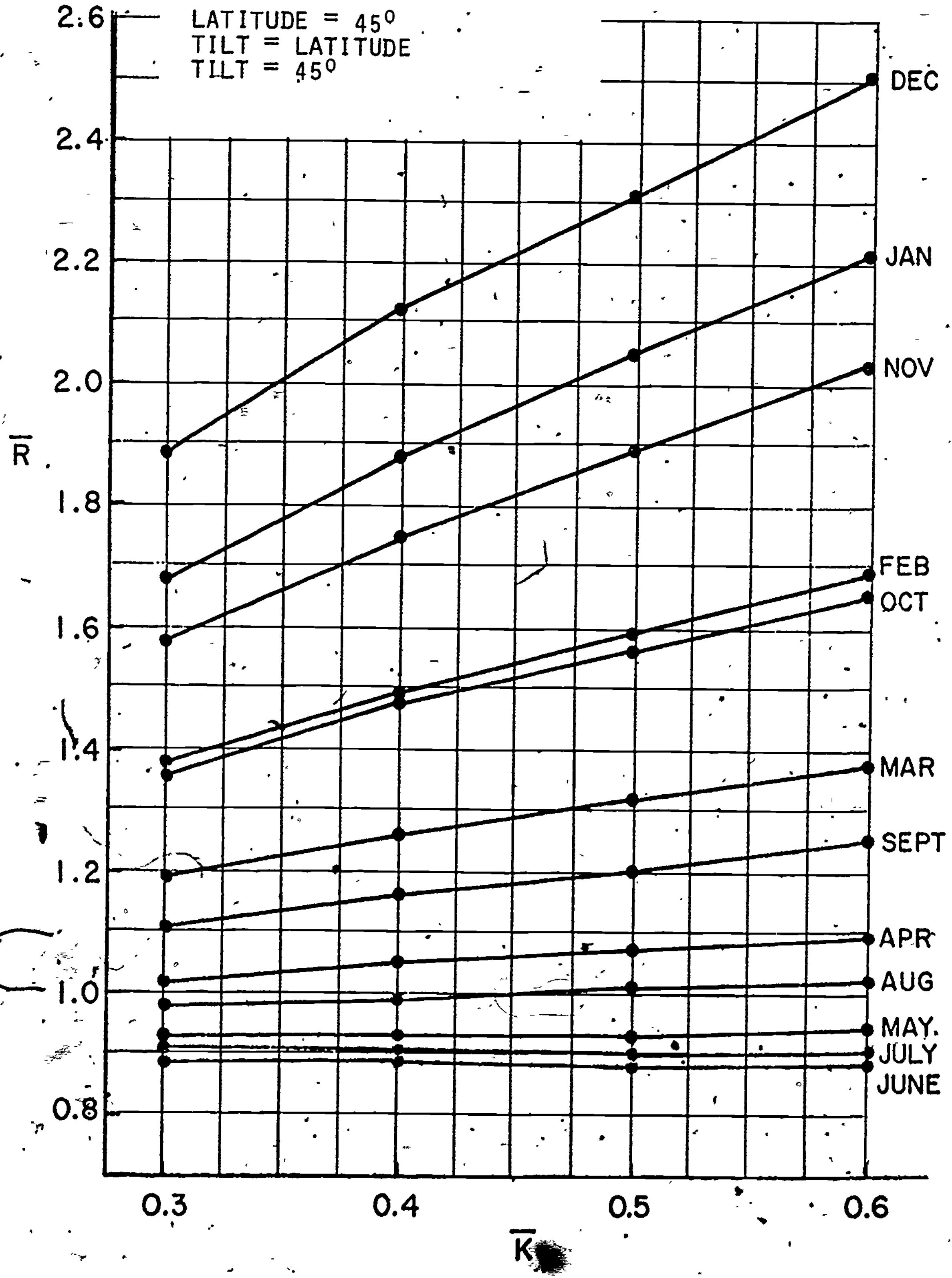
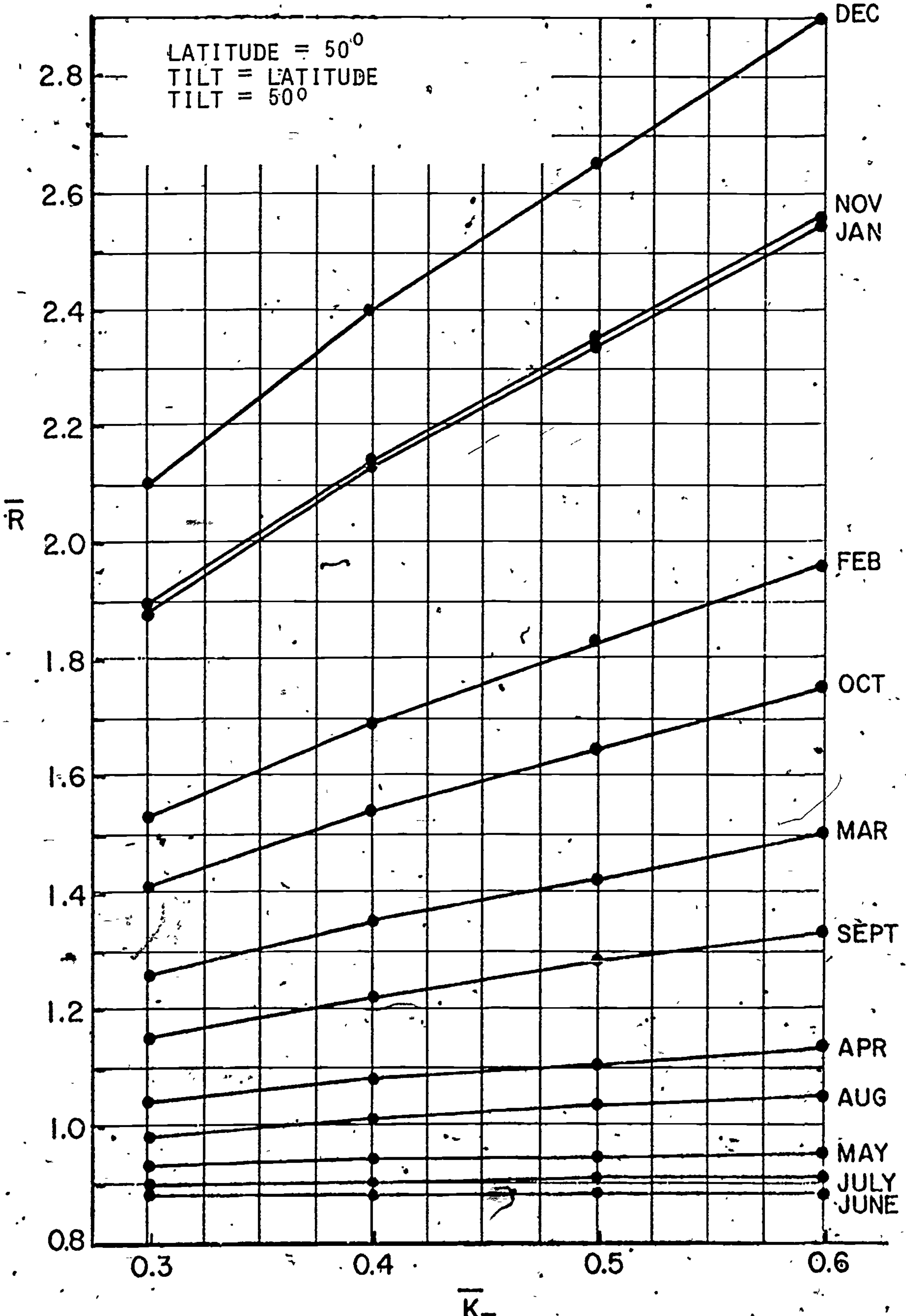


Figure 3-27
$\overline{R}$ versus $\overline{K}_T$



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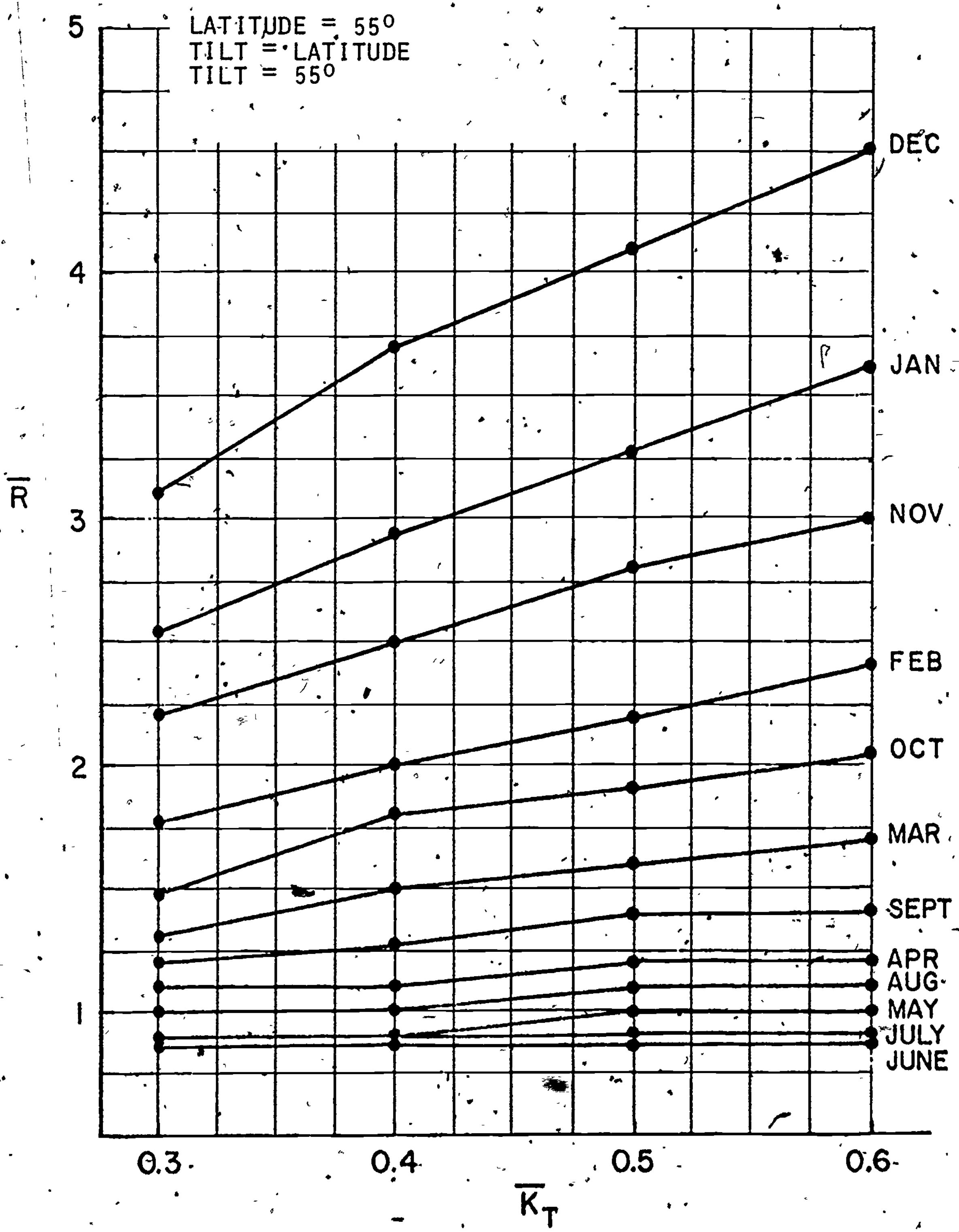

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$\overline{R}$ versus $\overline{K}_T$

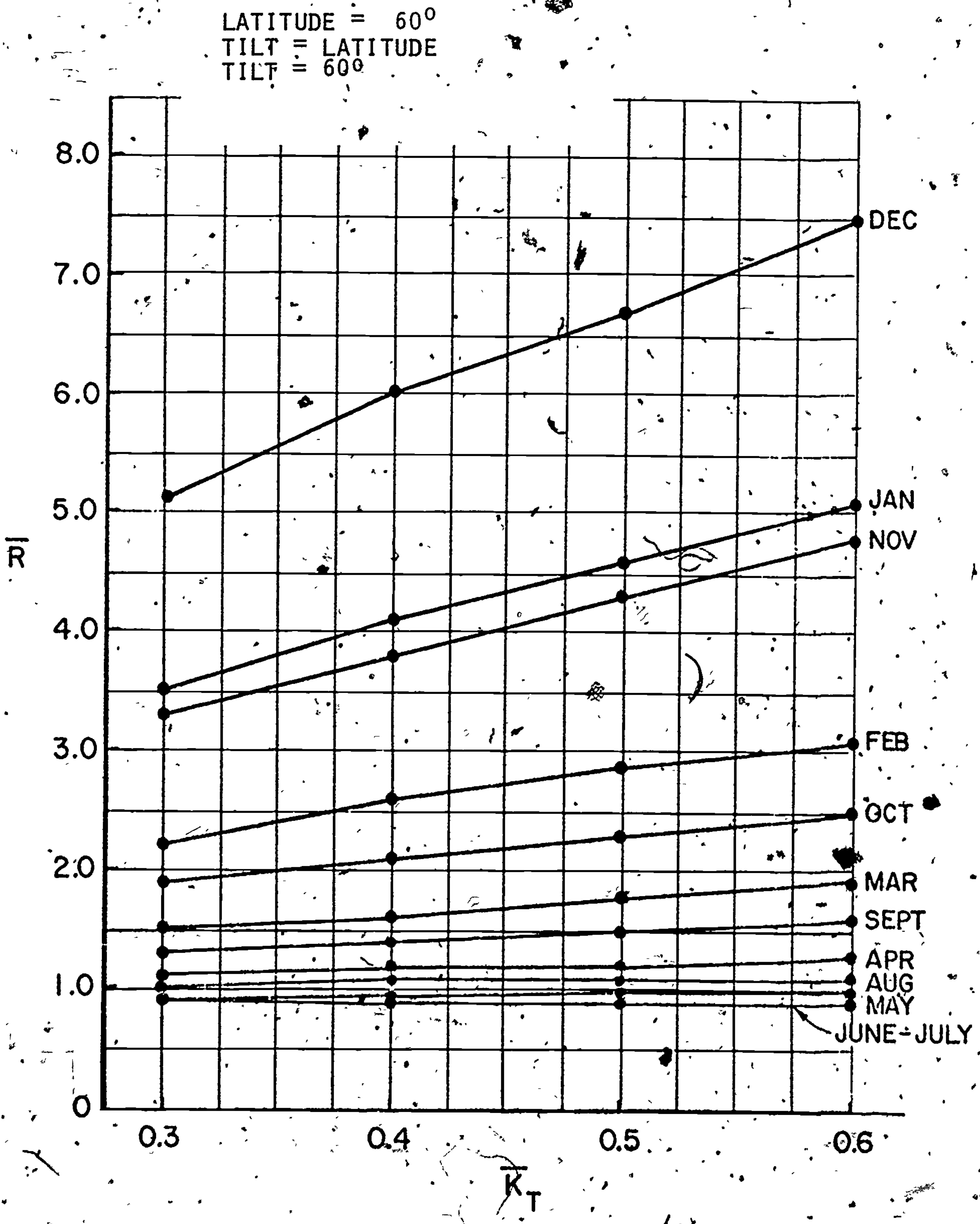


Figure 3-30
$\overline{R}$ versus $\overline{K}_T$

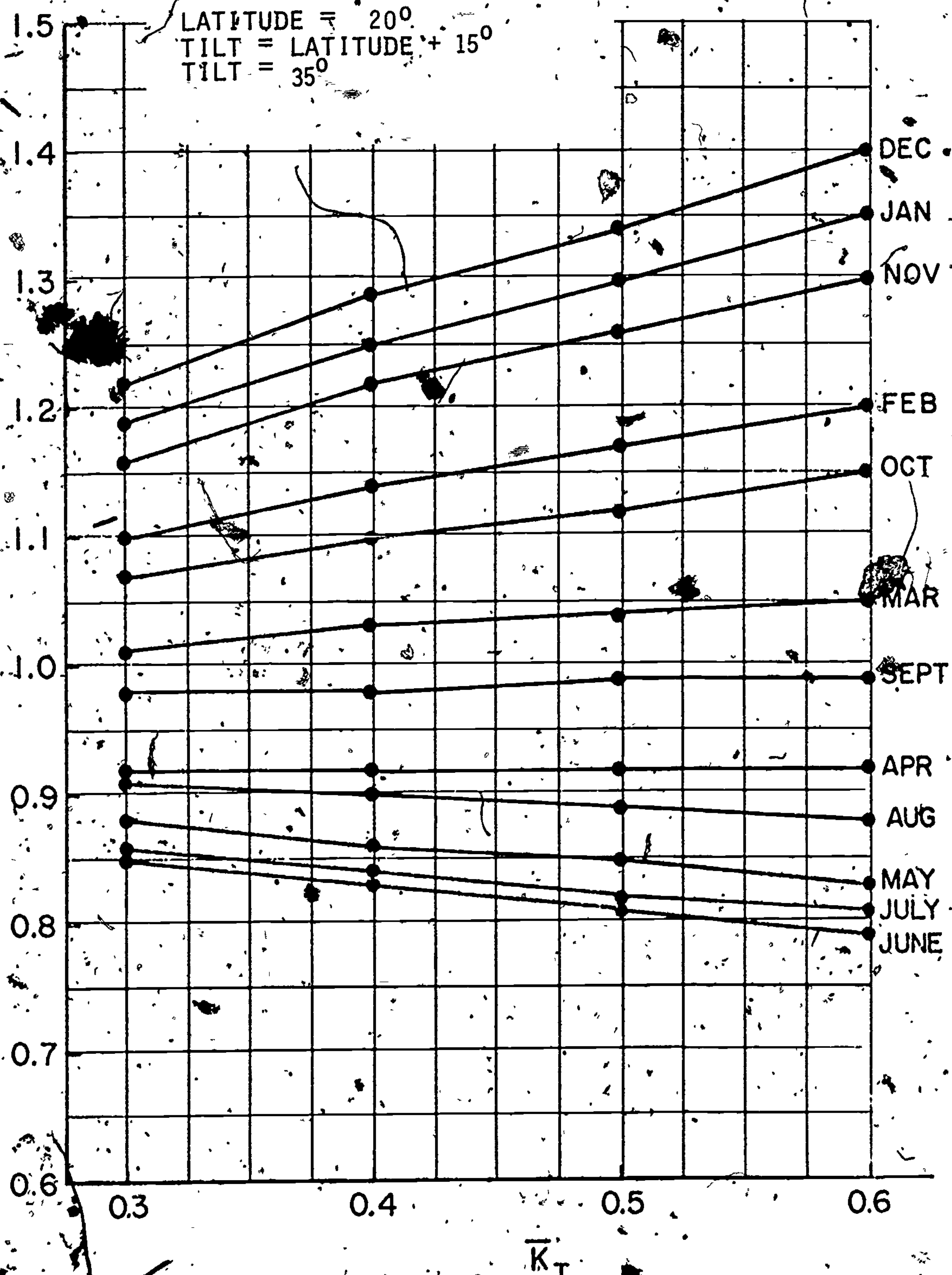
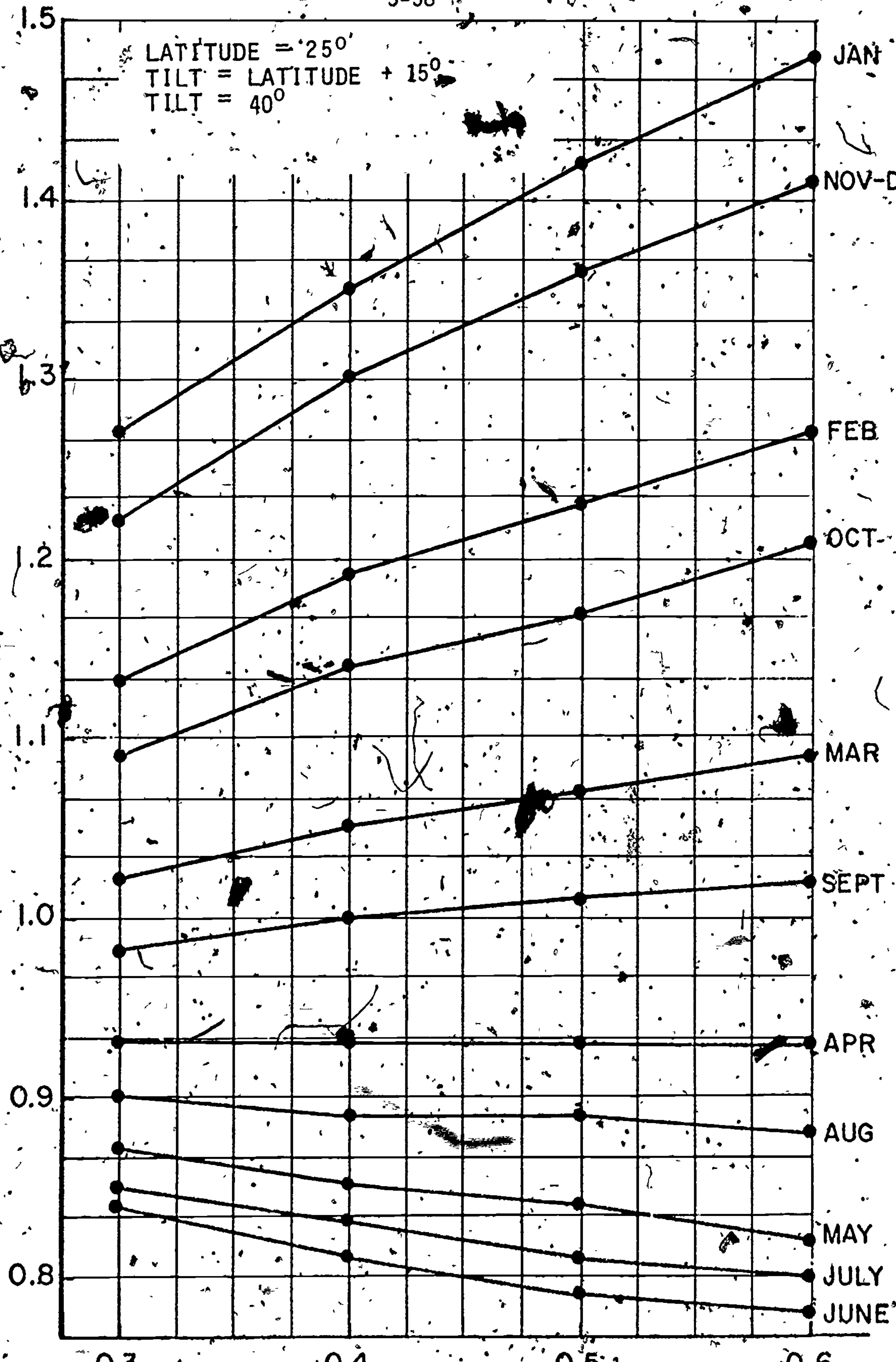


Figure 3-31

$\overline{R}$ versus $\overline{K}_T$



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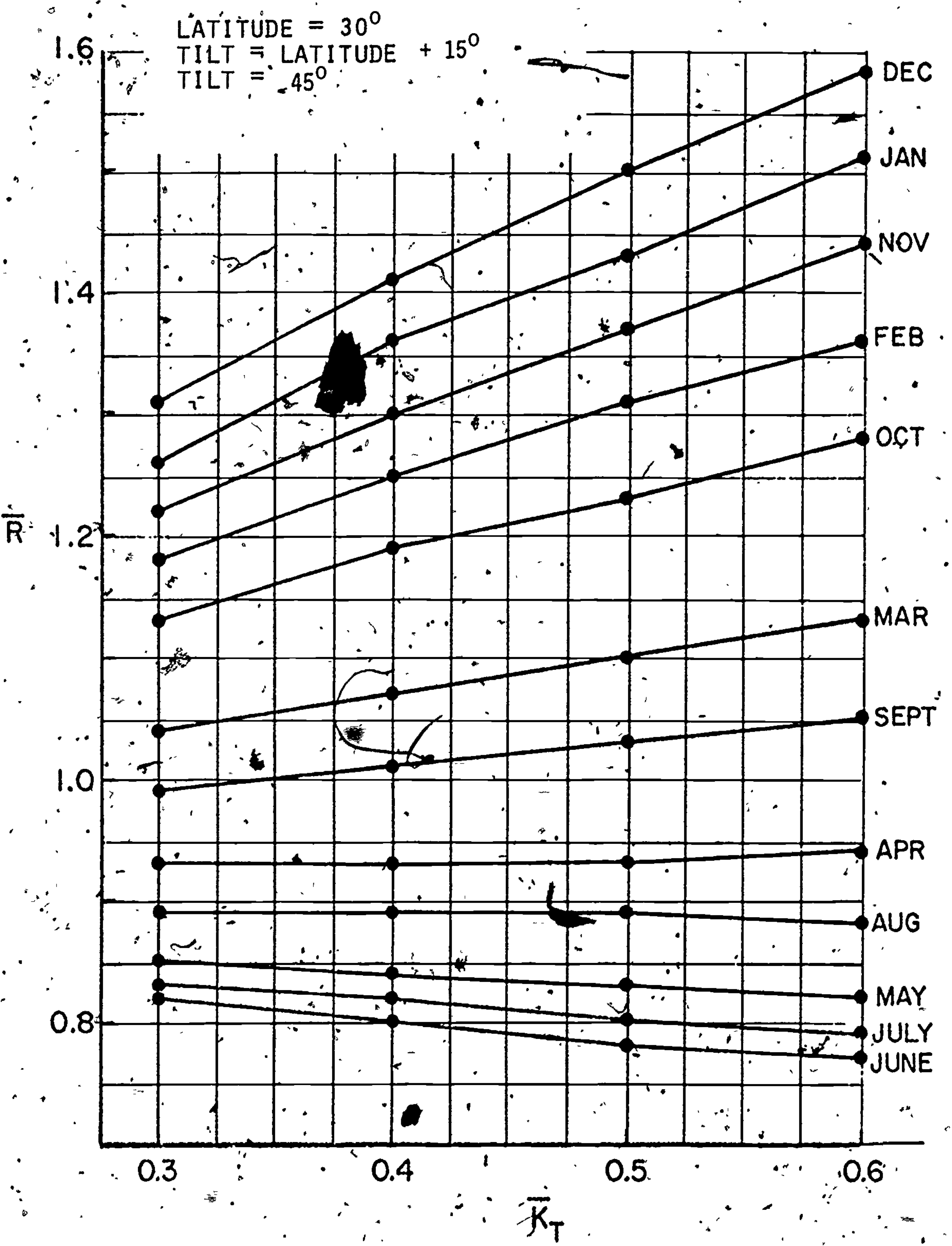
$\overline{R}$ versus $\overline{K}_T$



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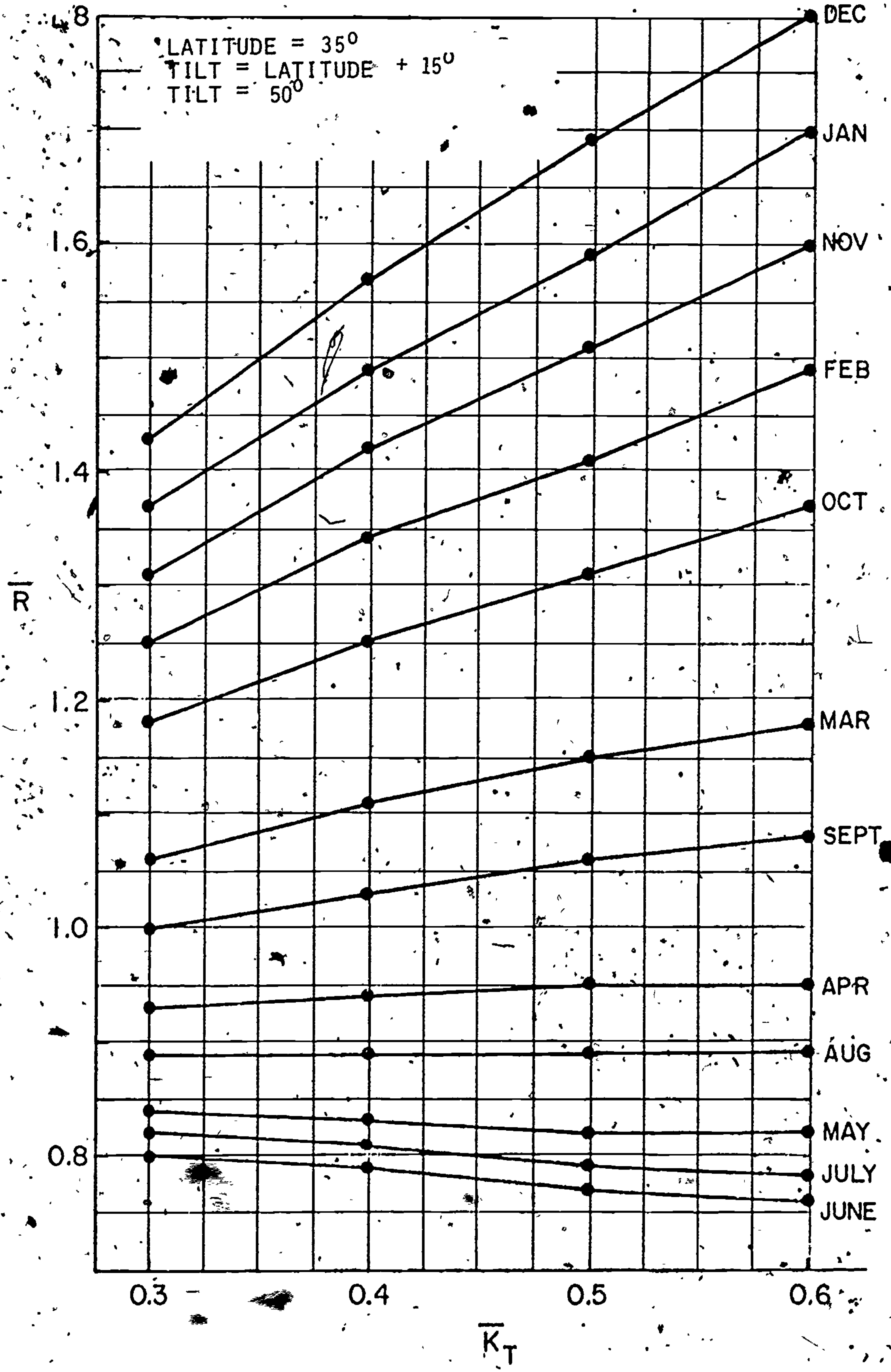

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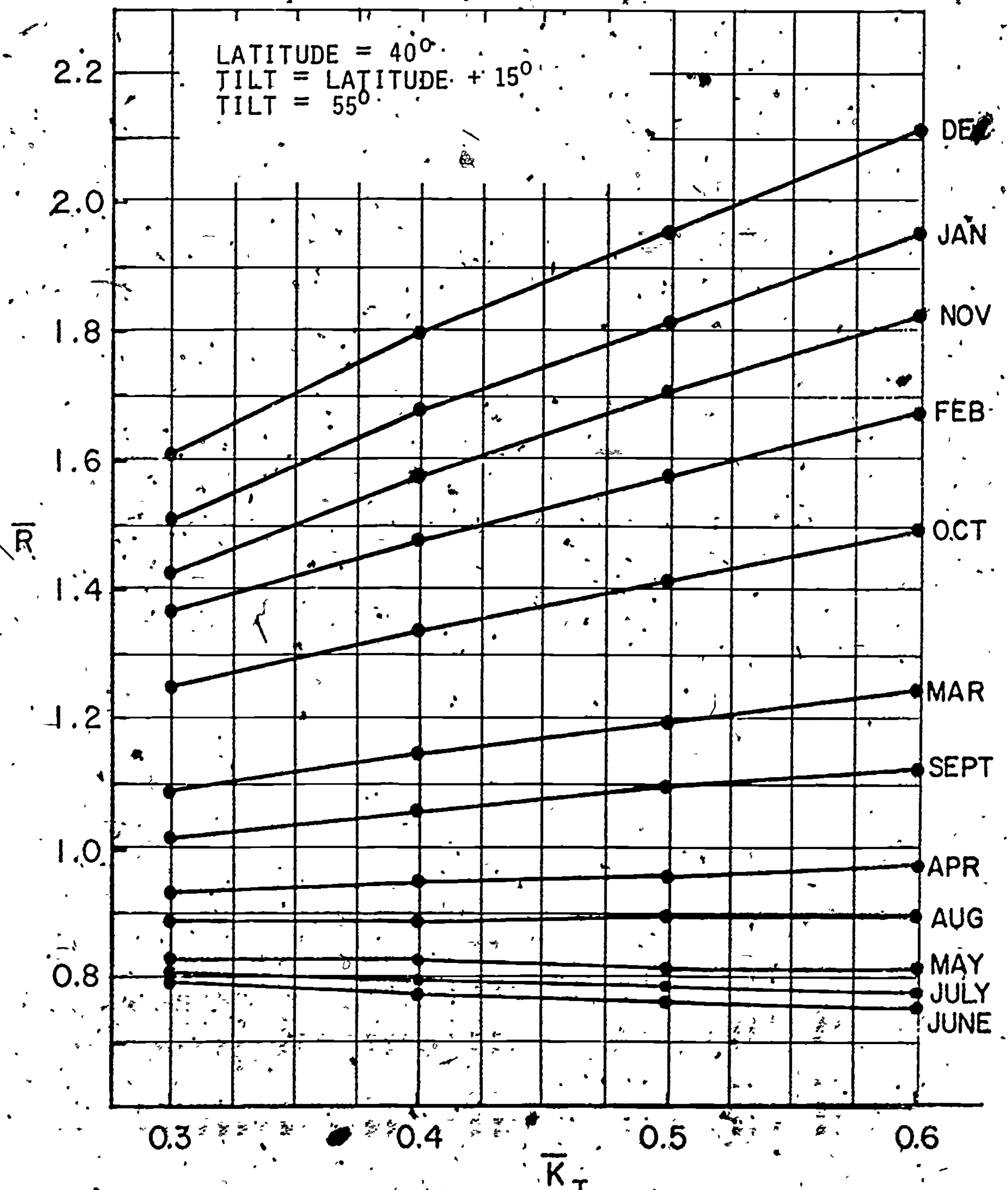
$\bar{R}$ versus $\bar{K}_T$



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$\overline{R}$ versus $\overline{K}_T$

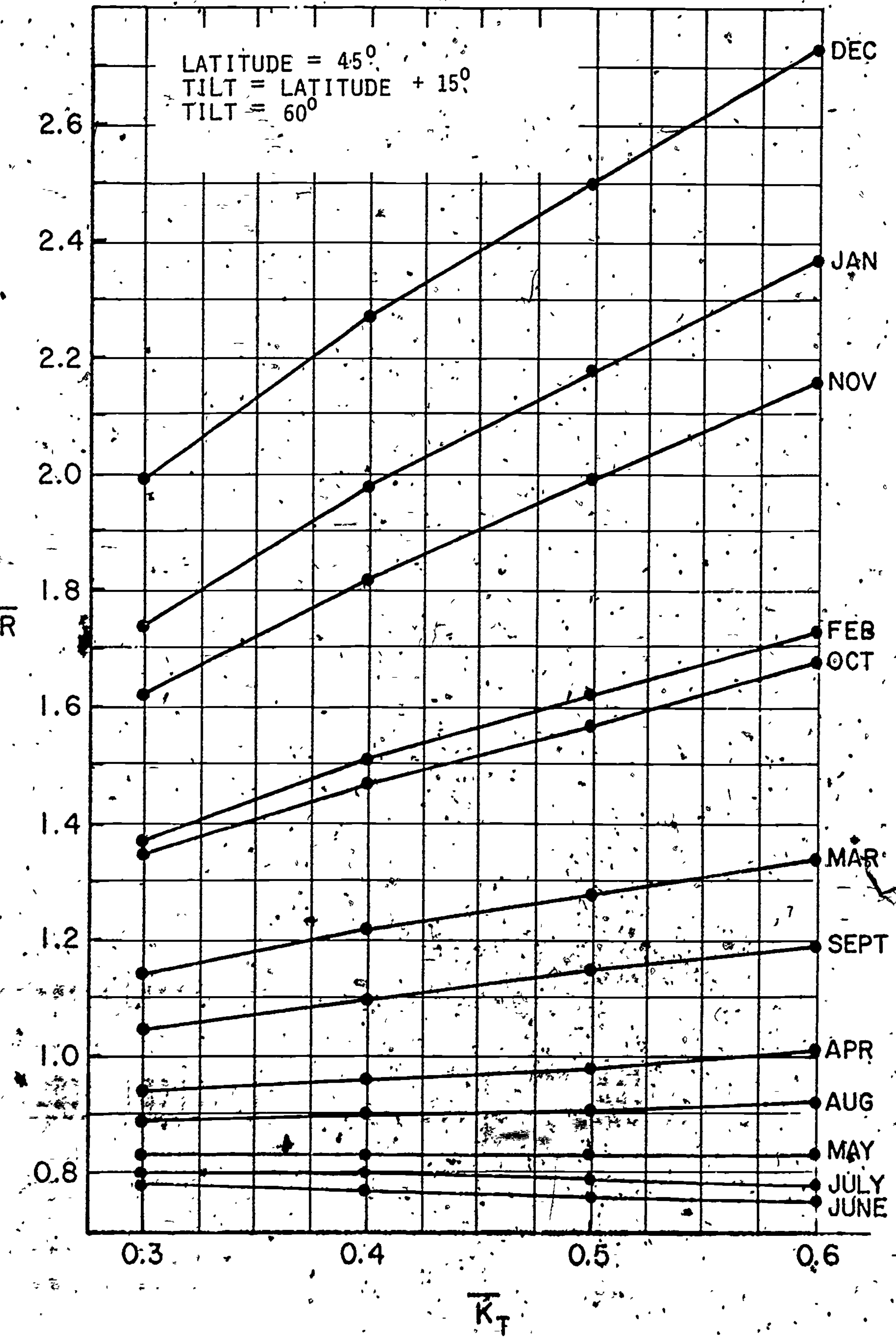

Figure 3-36
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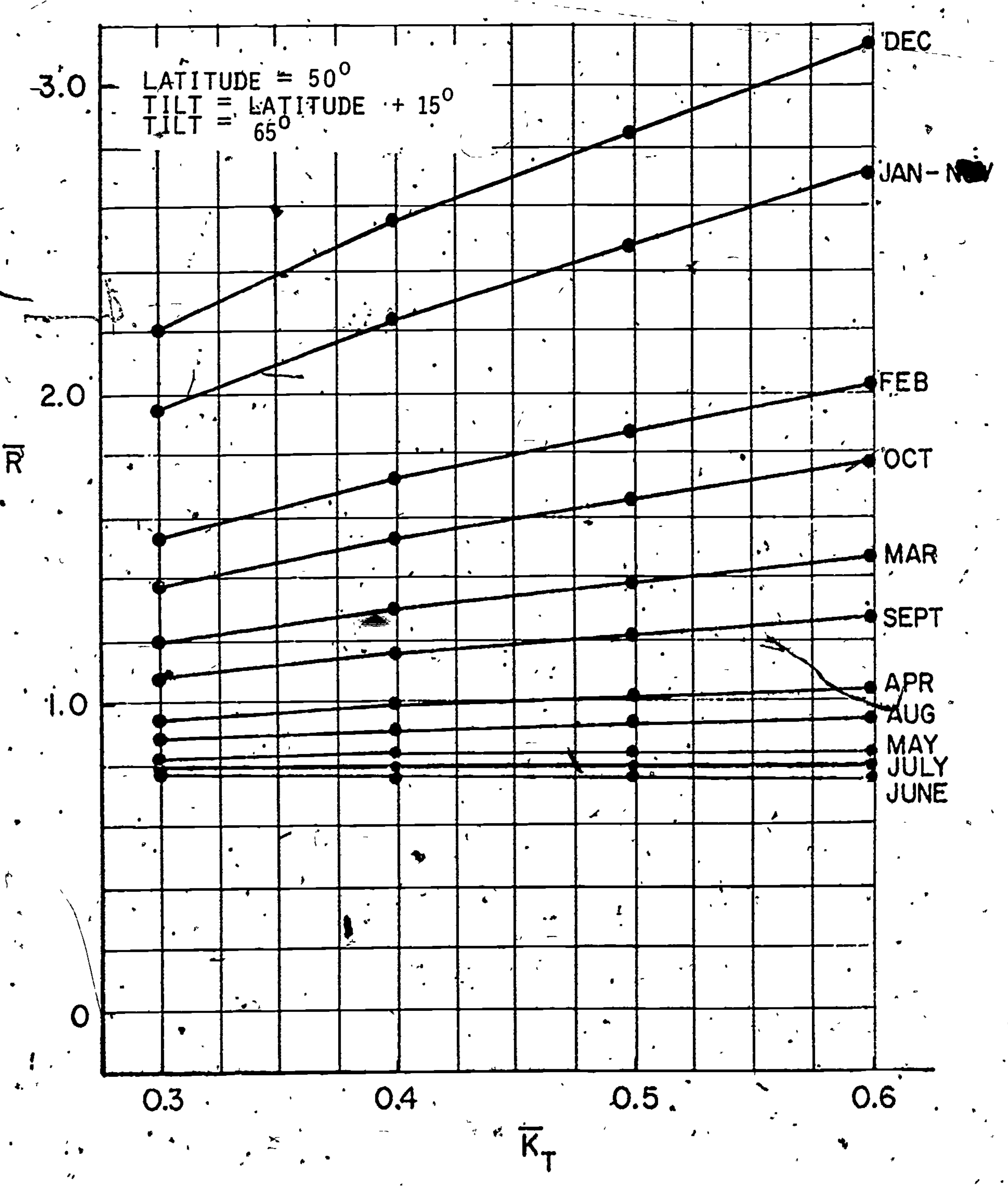

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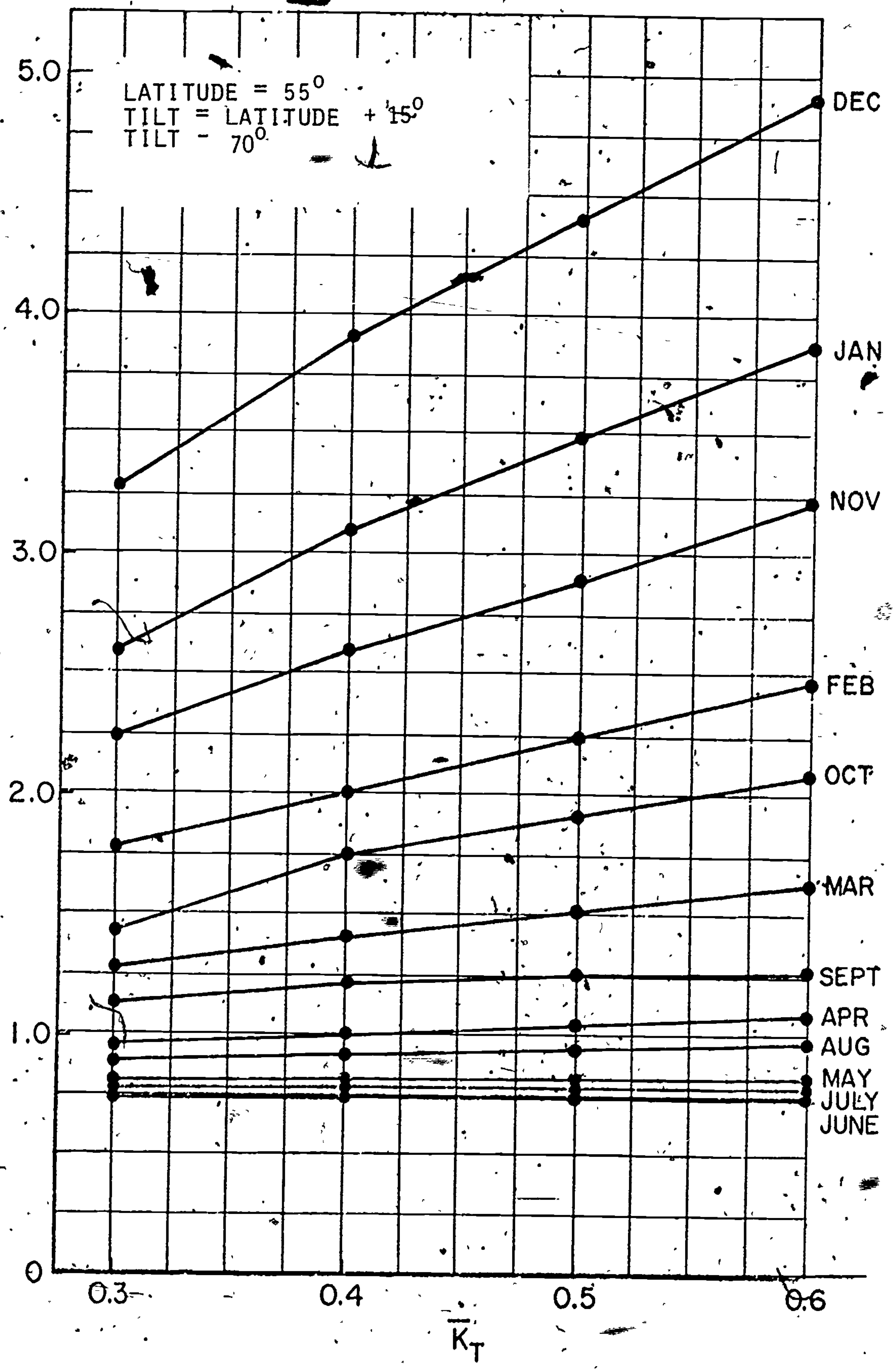

Figure 3-38

$\overline{R}$ versus $\overline{K}_T$

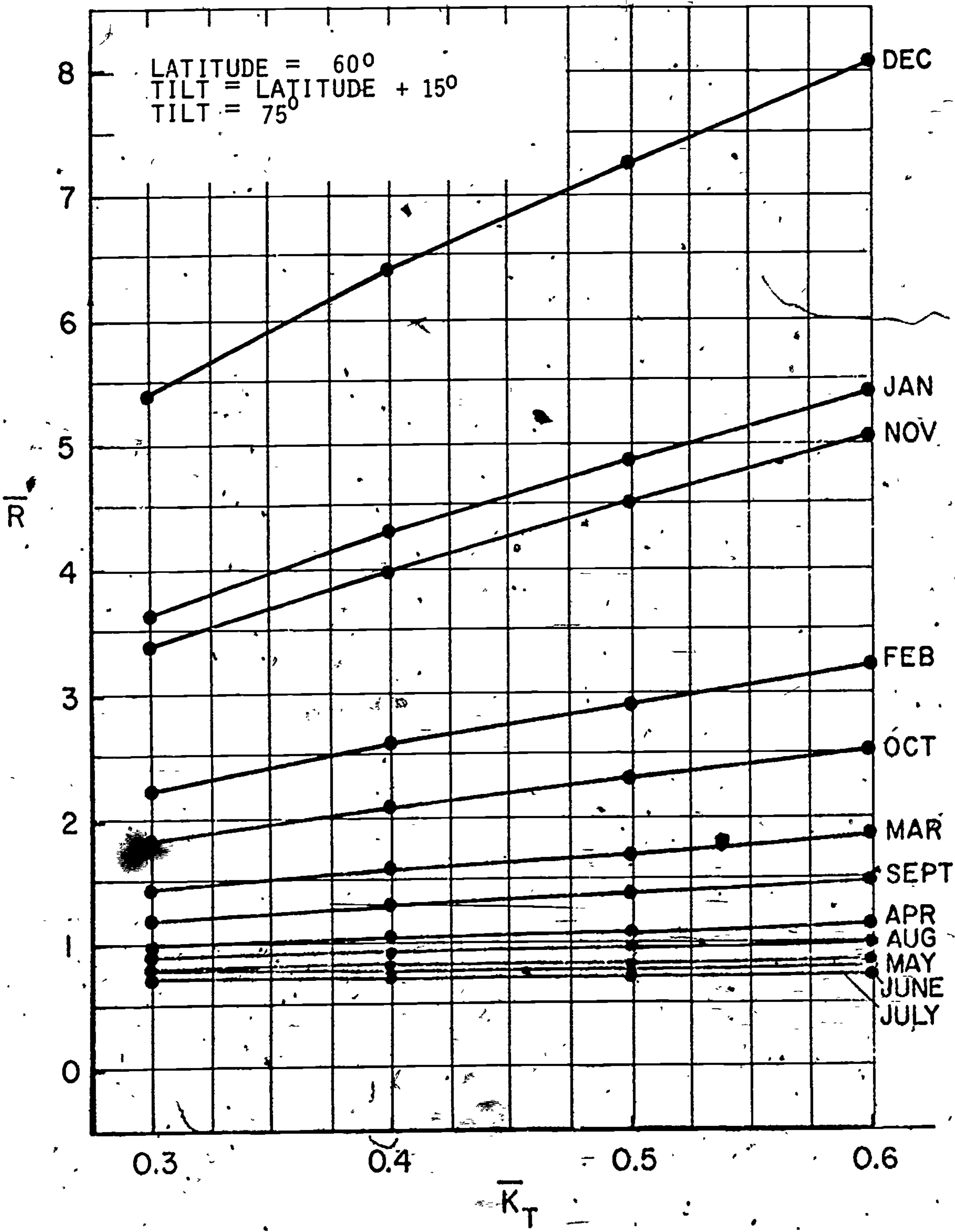

Figure 3-39

$\overline{R}$ versus $\overline{K}_T$

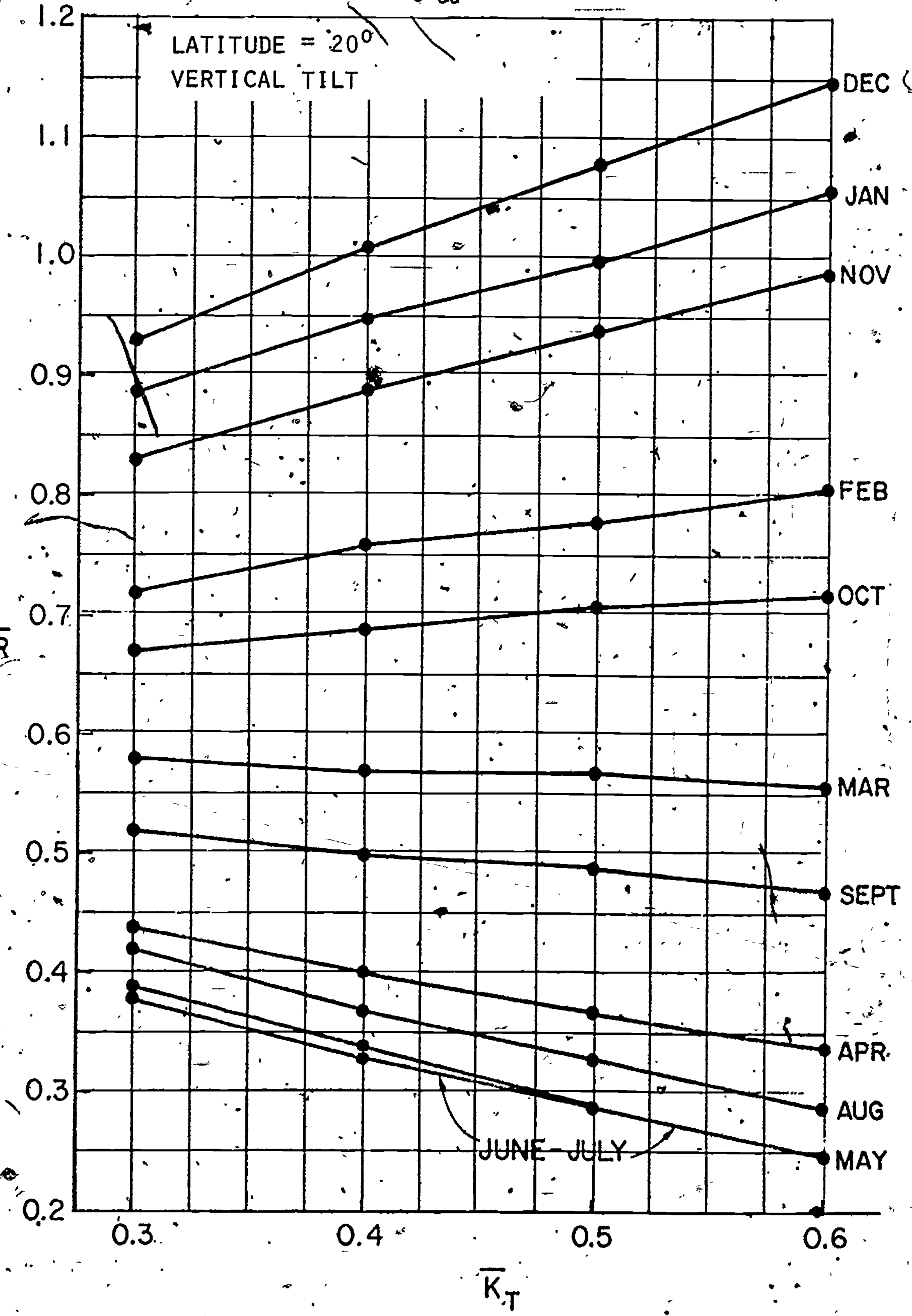

Figure 3-40
$\overline{R}$ versus $\overline{K}_T$

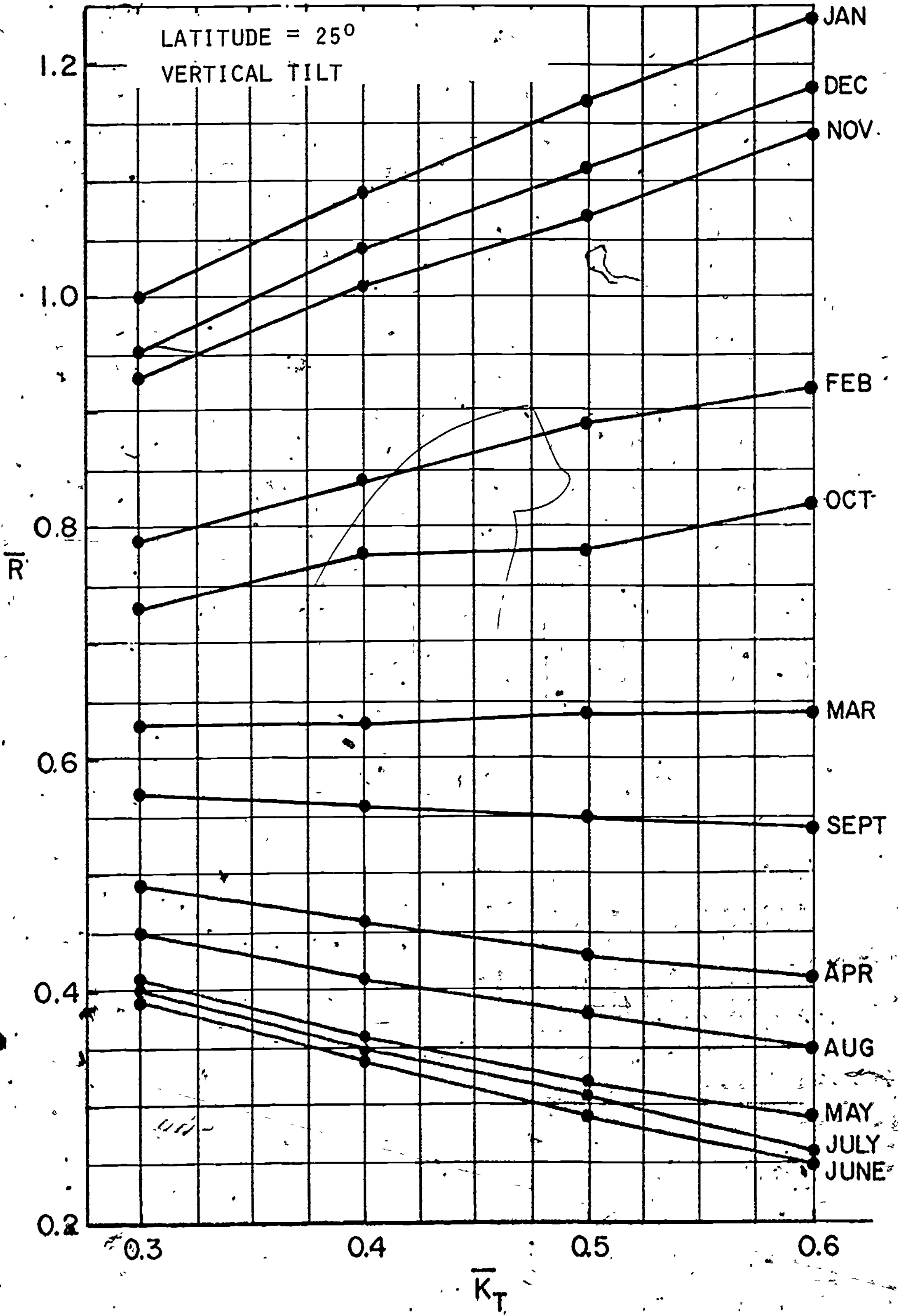

Figure 3-41

$\overline{R}$ versus $\overline{K_T}$

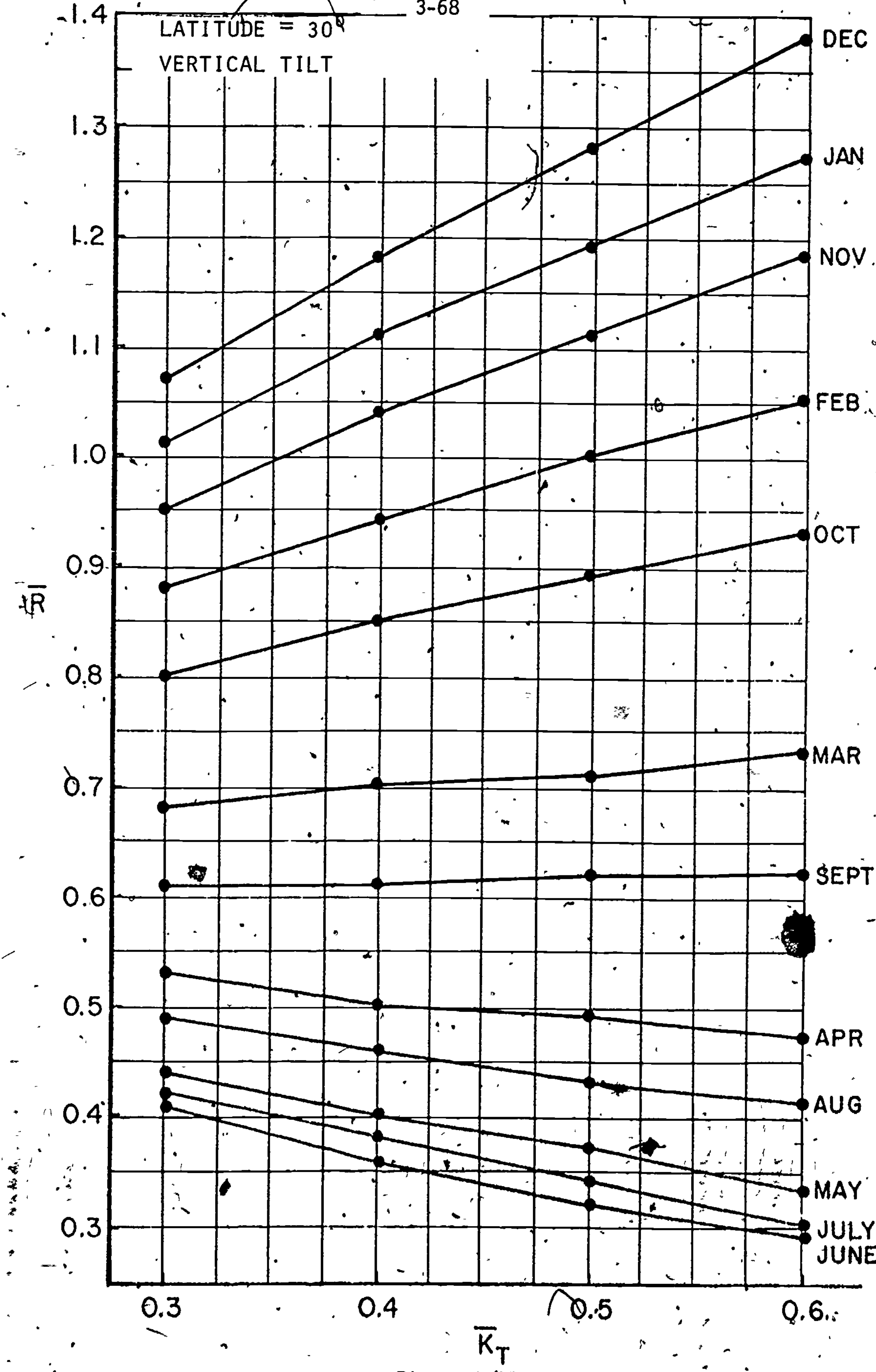

Figure 3-42
$\overline{R}$ versus $\overline{K}_T$

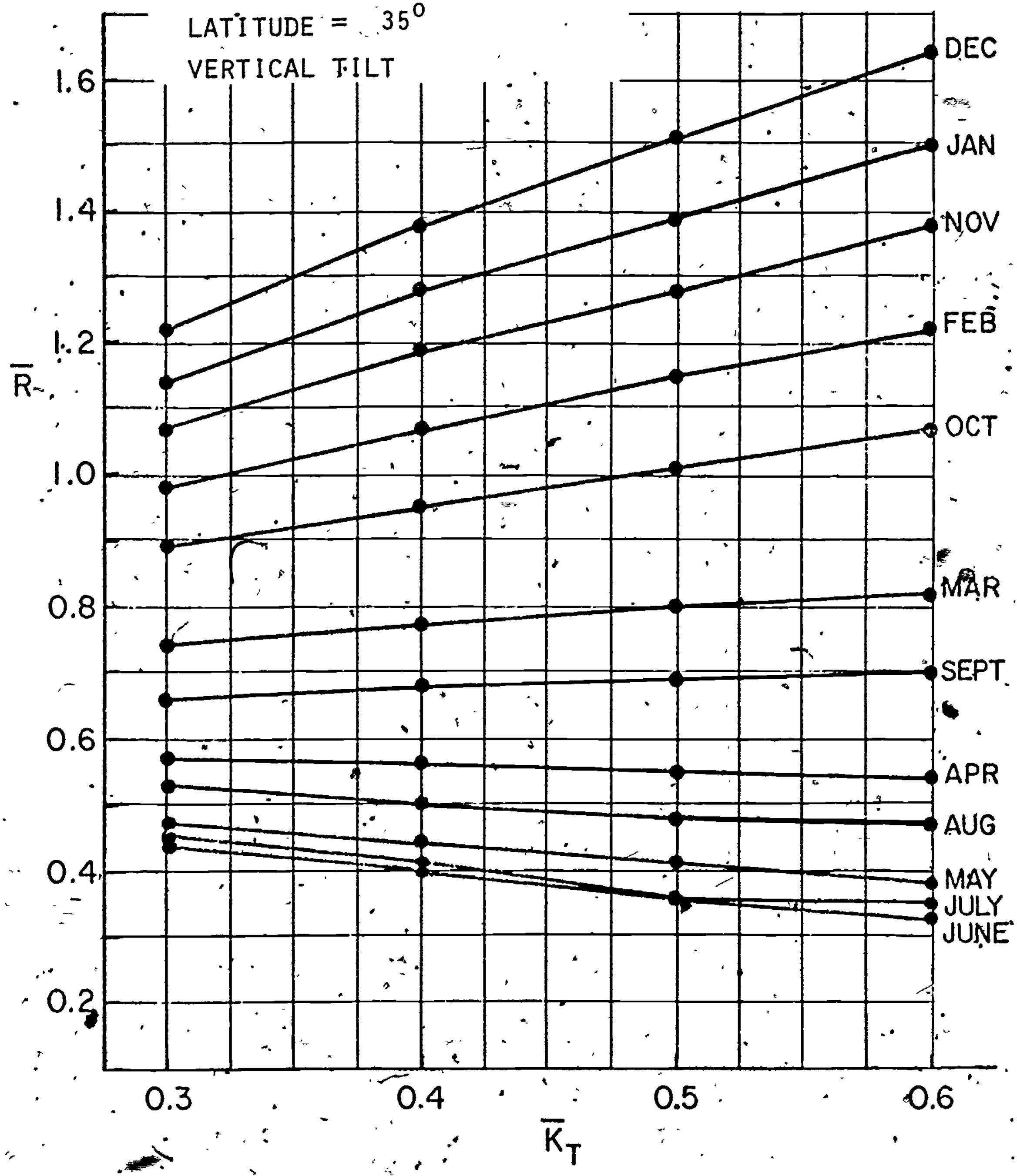

Figure 3-43

$\overline{R}$ versus $\overline{K}_T$

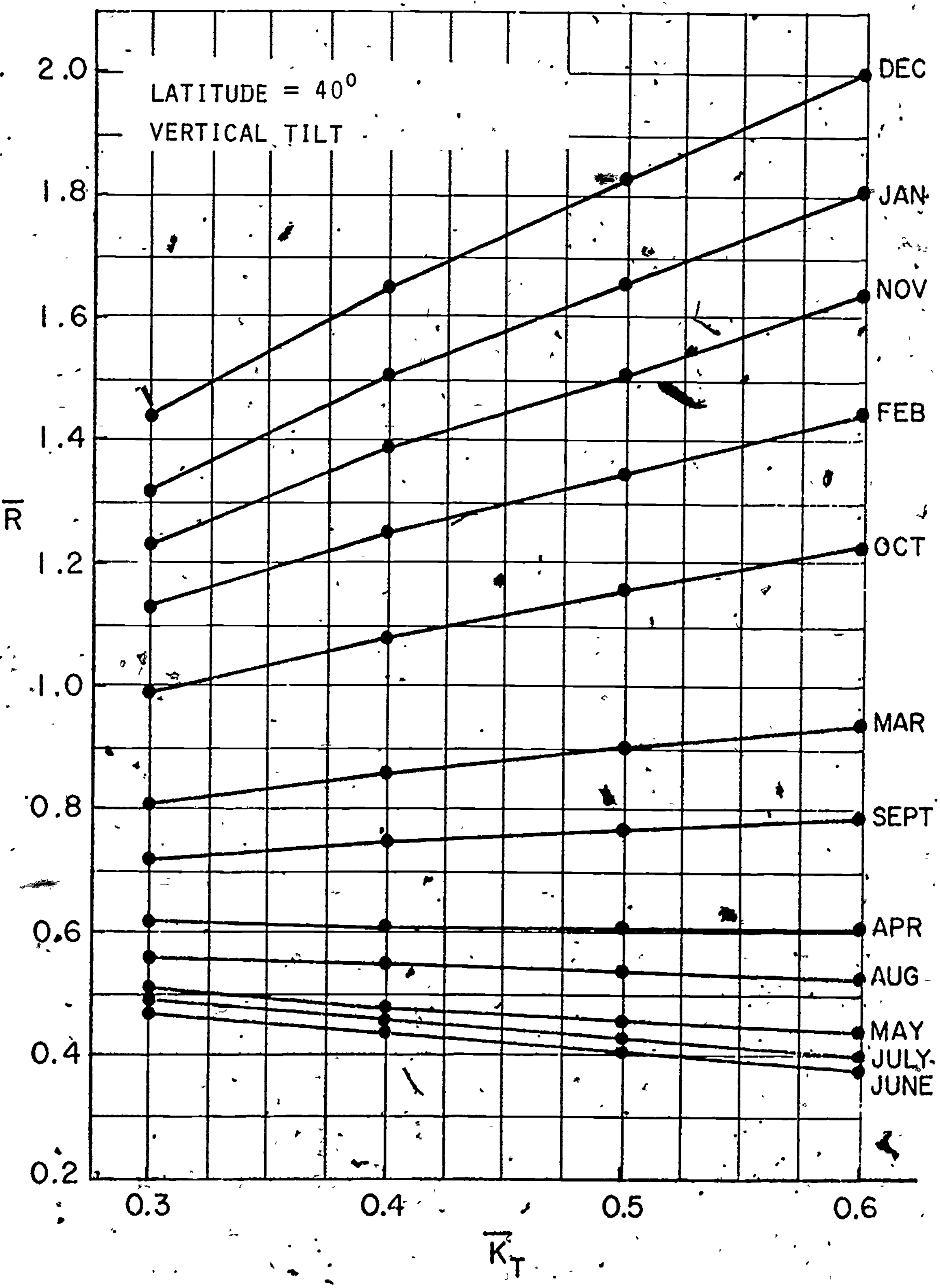

Figure 3-44
$\overline{R}$ versus $\overline{K}_T$

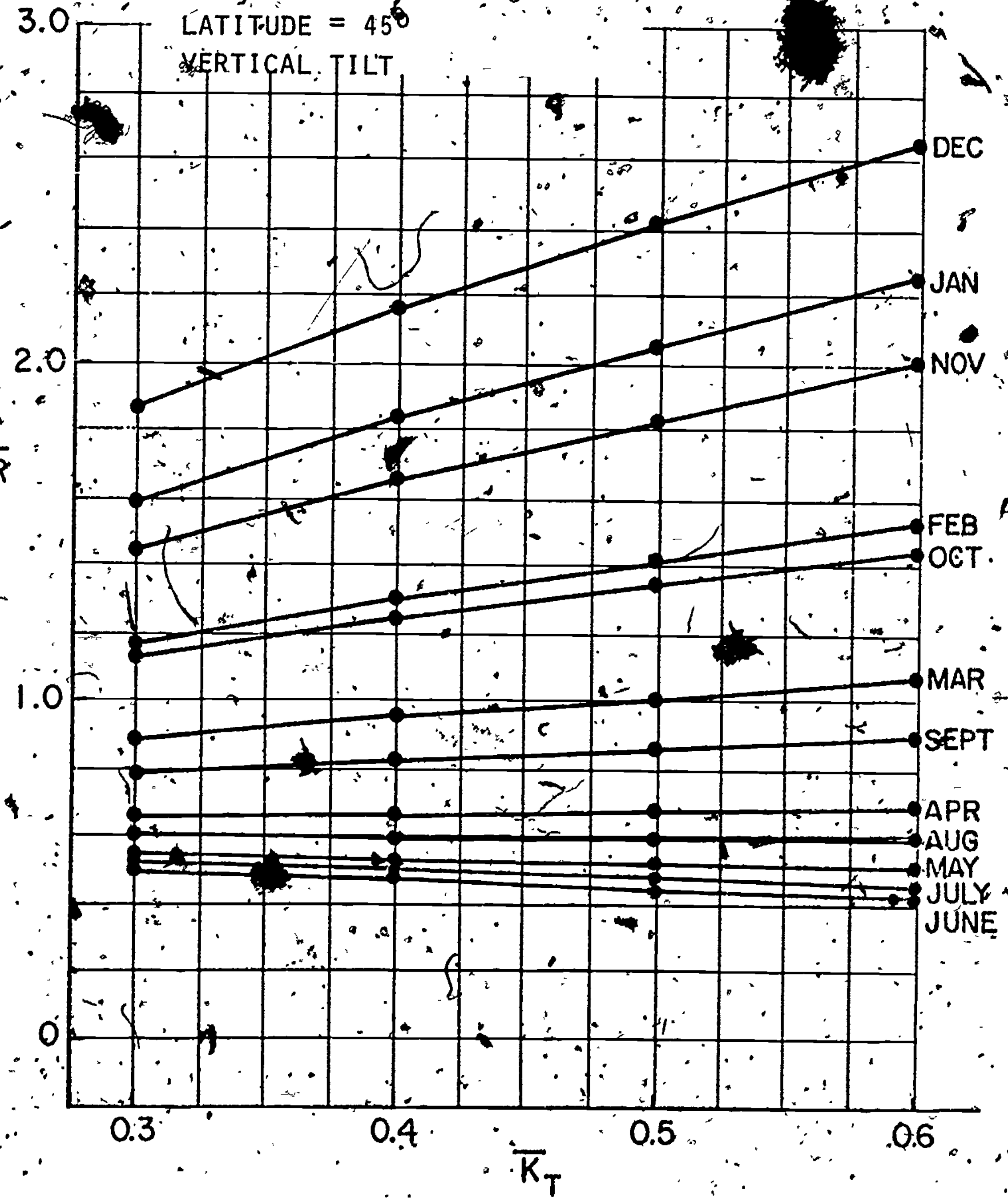

Figure 3-45
$\overline{R}$ versus $\overline{K}_T$

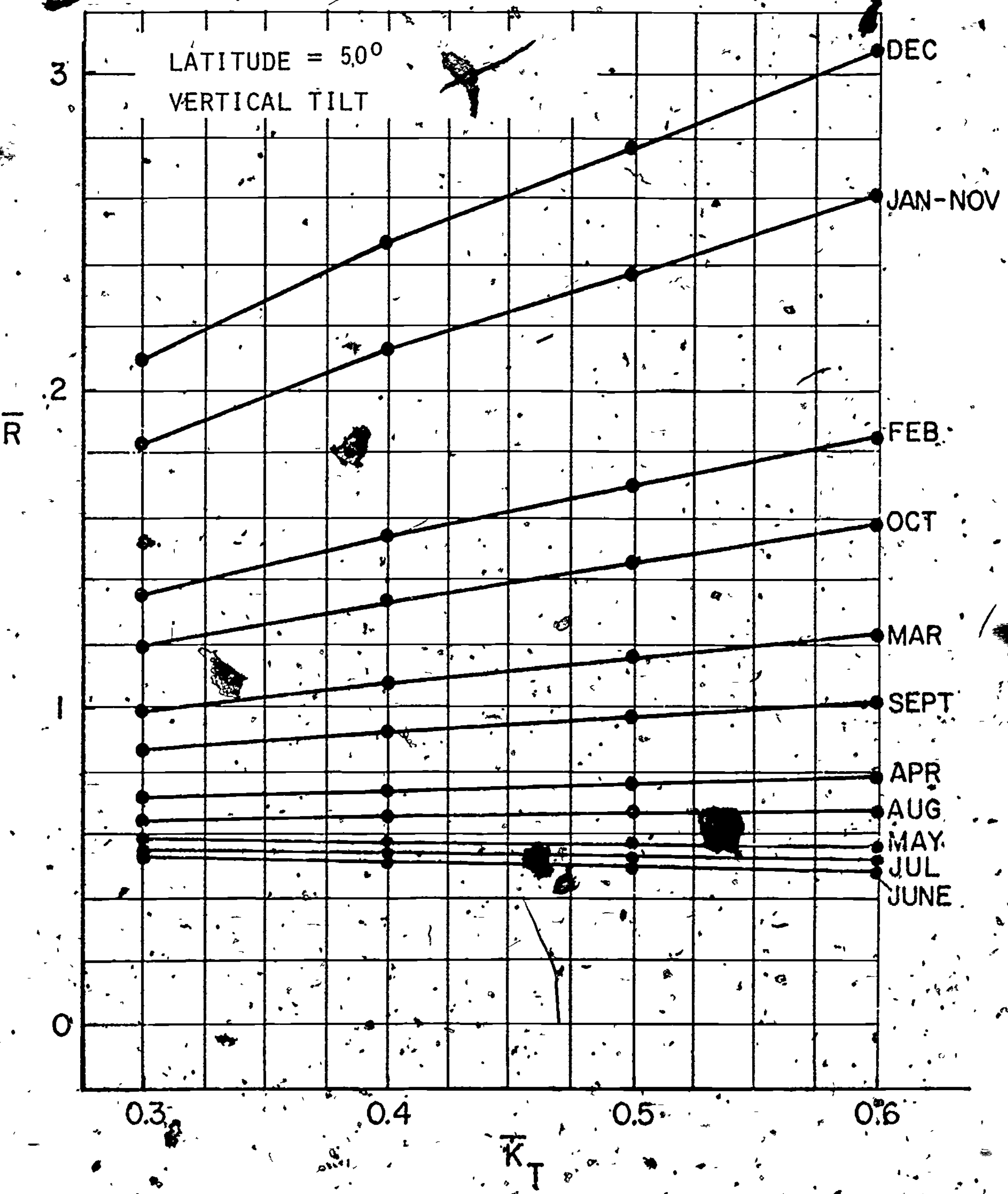

Figure 3-46
$\overline{R}$ versus $\overline{K}_T$

TRAINING COURSE IN
THE PRACTICAL ASPECTS OF
DESIGN OF SOLAR HEATING AND COOLING SYSTEMS
FOR
RESIDENTIAL BUILDINGS

MODULE 4

SYSTEM DESIGN GUIDELINES

SOLAR ENERGY APPLICATIONS LABORATORY
COLORADO STATE UNIVERSITY
FORT COLLINS, COLORADO

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GLOSSARY OF TERMS

ASHRAE	American Society of Heating, Refrigeration and Air Conditioning Engineers
Algorithm	A procedure, specifically for computations
Ambient	Prevailing environmental condition
Insolation	Solar radiation
Langley	Measure of solar radiation, 1 cal/(cm)^2, or 3.69 Btu/(ft^2)

LIST OF SYMBOLS

A	Collector area, ft^2
DD	Degree F-day $[65 - \frac{(T_{high} + T_{low})}{2}]$
$\overline{H}$	Total average daily solar radiation on a horizontal surface for a month (Btu/ft^2-day)
$\overline{H}_0$	Average daily extraterrestrial solar radiation on a horizontal surface for a month, $Btu/(ft^2$-day)
$\overline{H}_T$	Total average solar daily radiation on a tilted surface, (Btu/ft^2-day)
$\overline{K}_T$	Ratio $\overline{H}/\overline{H}_0$
L	Annual or monthly heat load, Btu
Q	Heat load, Btu/hr or Btu/DD
Q_{DES}	Heat requirement of building, Btu/DD
$\overline{R}$	Ratio $\overline{H}_T/\overline{H}$
T_{high}	High temperature for day
T_{low}	Low temperature for day

INTRODUCTION

TRAINEE-ORIENTED OBJECTIVE

The primary objective for the trainee in this module is to learn simple and rapid methods for sizing solar collectors and designing heating and service hot water systems.

SUB-OBJECTIVES

At the end of this module the trainee should be able to:

1. Relate information on solar input radiation to sizing systems.
2. Describe building load requirements.
3. Describe the efficiencies of components and systems.
4. Perform quick calculations.

It is important that the designer of solar heating and/or cooling systems be able to perform rapid and approximate calculations for determining the size of the various components that constitute a solar heating and/or cooling system. These approximation techniques presented in this module may be used to provide quick checks against more detailed analyses to ensure against possible gross errors, and may also be used to provide information to a potential client. From an approximate collector area sizing, and using an installed system cost based on collector area (say $20 per square foot of collector as given in Module 2), an approximate system cost can be estimated for the purpose of discussions with clients. It should be emphasized that the methods presented in this module are only approximate and should not be used for the detailed design of a system.

APPROXIMATE COLLECTOR SIZING METHODS

A. HUCK-WINN METHOD[1]

Design curves have been developed for air and water residential solar heating systems for space and service hot water heating, and service hot water heating separately. The system configurations modeled are shown in Figures 4-1 through 4-4, and the assumed daily hot water usage schedule is given in Table 4-1.

Several methods of plotting the annual fraction of the heating load supplied by the solar system, f, as a function of climatic conditions were tested. Of those methods tested, the parametric group $\overline{H}_T A/L$ gave the best results; $\overline{H}_T$ is the January solar insolation on the tilted collector surface, A is the collector area, and L is the total heating load for January.

Thirty cities, dispersed throughout the continental United States, were selected for the development of the design curves. The data points for each location were obtained by assuming a particular collector and collector area, and calculating the annual fraction of the heating load supplied by the solar system for the given area.

Three restrictions were imposed for the development of the curves. One, the annual fraction of the load supplied by the solar system, f, must be at least 0.1. Two, f could not exceed 0.9. And three, the collector area was not to exceed 1200 square feet. These restrictions

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[1] Steven Huck and C. Byron Winn, "Design Charts for Residential Solar Heating Systems," submitted to Solar Energy, December 1976.

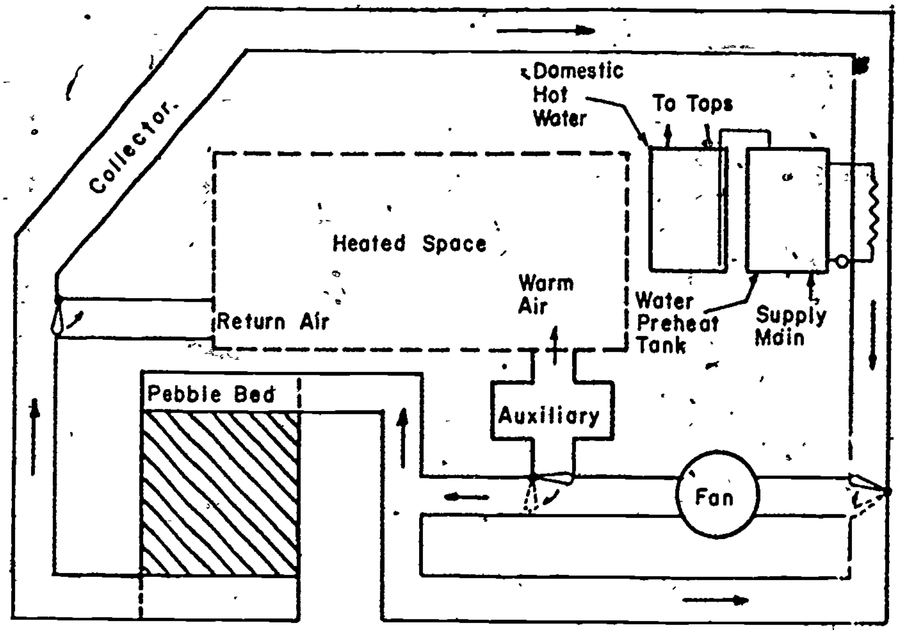
Collector
Domestic Hot Water
To Taps
Heated Space
Warm Air
Return Air
Water Preheat Tank
Supply Main
Pebble Bed
Auxiliary
Fan

Figure 4-1. Air System - Space and Service Hot Water Heating

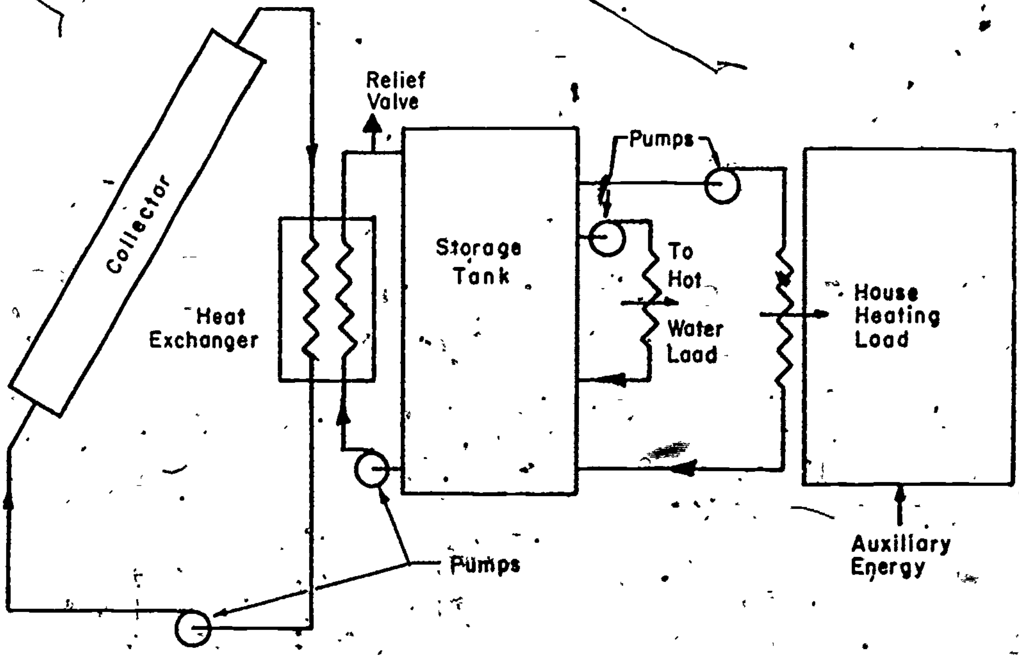
Collector
Relief Valve
Pumps
Heat Exchanger
Storage Tank
To Hot Water Load
House Heating Load
Pumps
Auxiliary Energy

Figure 4-2. Water System - Space and Service Hot Water Heating

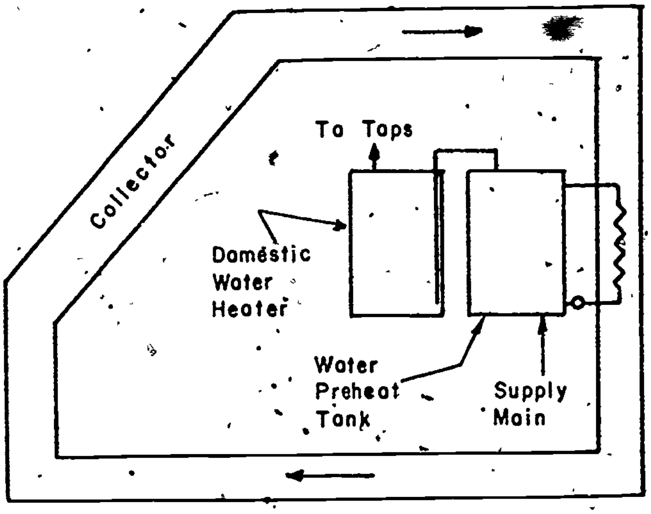
To Taps
Collector
Domestic
Water
Heater
Water
Preheat
Tank
Supply
Main

Figure 4-3. Air System - Service Hot Water Heating

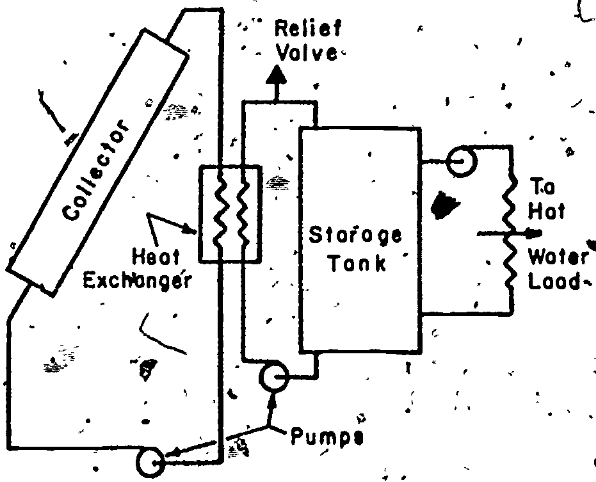
Relief
Valve
Collector
Heat
Exchanger
Storage
Tank
To
Hot
Water
Load
Pumps

Figure 4-4. Water System - Service Hot Water Heating

Table 4-1. Schedule of Daily Hot Water Usage

Time	Percent of Total Usage	Time	Percent of Total Usage
Midnight - 1	2.25	Noon - 1	3.60
1 - 2	0.00	1 - 2	5.10
2 - 3	0.00	2 - 3	2.70
3 - 4	0.00	3 - 4	2.40
4 - 5	0.00	4 - 5	2.10
5 - 6	0.00	5 - 6	3.75
6 - 7	1.50	6 - 7	6.75
7 - 8	4.65	7 - 8	11.60
8 - 9	7.25	8 - 9	9.60
9 - 10	8.49	9 - 10	6.90
10 - 11	6.90	10 - 11	5.46
11 - Noon	4.50	11 - Midnight	4.65

were imposed for economic reasons. A system designed to supply less than 10 percent of the required load would probably never realize a dollar savings due to the high fixed costs of the system. These fixed costs would be for required heat exchangers, pumps or blowers, and piping or ducts. On the other hand, a system designed to supply more than 90 percent of the required load may not realize a dollar savings due to the high cost of the collector array that would be required. For this same reason the collector array area was not allowed to exceed the 1200-square-foot limitation. These are reasonable restrictions for residential construction.

A least-squares, polynomial curve fit of the form

$$f = \beta_0 + \beta_1 \bar{H}_T A/L + \beta_2 (\bar{H}_T A/L)^2 + \ldots + \beta_N (\bar{H}_T A/L)^N$$

was selected to fit the data generated for the three design cases, for all of the collectors tested. The design curves given in Figures 4-5 through 4-7 show the results of the curve fitting process for each collector analyzed.* Table 4-2 outlines the resulting standard deviations for each design case and collector. The values that were assumed for the collector parameters and their respective code numbers, for use with the design curves, are given in Table 4-3. Values for additional collectors are shown in Table 4-4.

The following example problem illustrates the use of these curves. Suppose that we wish to design a solar system to provide approximately 75 percent of the space heating and service hot water requirements for a house to be located near Denver, Colorado. The house has a heat load of 24,000 Btu/DD (including service hot water).

The January heating requirements are determined by multiplying the heat load by the average number of degree days for January as determined from Table 4-5, which gives normal heating degree days for many locations in the United States.** A degree day is defined as the difference between the average temperature and 65°F. That is, if the average temperature for a day is 30°F, then there would be 65 - 30 = 35 degree days in that day. Continuing with our example we obtain

$$L = (24{,}000\ \frac{\text{Btu}}{\text{DD}})\ (1{,}132\ \text{DD})$$

$$= 27.168 \times 10^{6}\ \text{Btu}\ ,$$

or 27.168 million Btu for the month of January.

The radiation $\overline{H}_T$ on the tilted surface is determined by the use of the material presented in Module 3. First we find the radiation on

---

* See Approximate Size Curves, pages 4-27 through 4-32

** See Degree Day Tables, pages 4-45 through 4-53

Table 4-2. Standard Deviations for Design Curves of Figures 4-5 Through 4-7

Case Description	Figure	Coll. No.	Std. Dev.
Air System - Space Heating and Service Hot Water Heating	4-5	7	0.03124
Water System - Space Heating and Service Hot Water Heating	4-6	1	0.02922
		2	0.02953
		3	0.03847
		4	0.02904
		5	0.03202
		6	0.03871
Air and Water Systems - Service Hot Water Heating	4-7	1	0.04376
		2	0.04249
		3	0.04425
		4	0.04416
		5	0.03910
		6	0.04896
		7	0.04222

Table 4-3. Solar Collector Data

Collector Number	Type	Manufacturer	$F_R\overline{\tau\alpha}$	F_RU_L Btu/hr-ft^2-F
1	Water	NASA/Honeywell	.70	.53
2	Water	NASA/MSFC	.53	.66
3	Water	NASA/Honeywell	.80	1.17
4	Water	NASA/Honeywell Mylar Honeycomb	.74	.55
5	Water	NASA/Honeywell	.71	.74
6	Water	PPG	.62	.94
7	Air	Solaron	.52	.52

Assumed F_R'/F_R = 0.97 for all water collectors
= 1.00 for air collectors (no heat exchanger)

Where F_R'/F_R is the heat exchanger factor
F_R' is the collector efficiency factor
F_R is the collector heat removal factor
U_L is the collector overall loss
$\tau\alpha$ is the collector transmittance-absorbance product

Table 4-4. Collector Parameters

Manufacturer	Type	$F_R\overline{\tau\alpha}$	F_RU_L (Btu/ft^{2}h^{o}F)
Owens-Illinois	Liquid	.447	.206
Intertechnology	Liquid	.650	.610
GE	Liquid	.639	.614
LeRC	Liquid	.745	.820
Rocky Mountain	Liquid	.679	.789
Southwest Std.	Liquid	.672	.794
Sunworks	Liquid	.650	.789
Trantor	Liquid	.700	.83*
Miromit	Liquid	.724	.947
Beasley	Liquid	.600	.759
Soltex	Liquid	.600	.759
Revere	Liquid	.716	.964
Barber	Liquid	.816	1.204
Solar Products	Liquid	.600	1.057

Taken from graphical presentation in "Comparison of Flat-Plate Collector Performance Obtained under Controlled Conditions in a Solar Simulator," by S. M. Johnson and F. F. Simon of Lewis Research Center, NASA, Cleveland, Ohio.

*Based on a linearized approximation of the efficiency curve.

a horizontal surface from Figure 3-1. We see that this is approximately 201 ly/day for the region near Denver, Colorado. This figure is converted to Btu/ft^2-day by multiplying by 3.69. Thus,

$$\overline{H} = (201\ \frac{\text{ly}}{\text{day}})\ (3.69\ \frac{\text{Btu/ft}^2}{\text{ly}})$$

$$= 742\ \text{Btu/ft}^2\text{-day}$$

The factor, $\overline{R}$, for converting this to the tilted surface may be determined from Figure 3-35 using 0.55 for $\overline{K}_T$, as determined by the average ratio between $\overline{H}_o$ and $\overline{H}$, where $\overline{H}_o$ may be determined from Table 3.2. The latitude is approximately 40^{o}N. We obtain

$$\overline{H}_o = 1326\ \text{Btu/ft}^2\text{-day}$$

and

$$\overline{K}_T = 742/1326 = 0.55.$$

Then,

$$\overline{R} = 1.89$$

and finally

$$\overline{H}_T = (1.89)(\overline{H})$$

$$= 1402\ \text{Btu/ft}^2\text{-day}$$

Then, from Figure 4-6, assuming we wish to use a water system with collector number 6, we obtain

$$\frac{\overline{H}_T A}{L} \approx 1.75$$

as the value required to supply approximately three-fourths of the heating requirements. The required collector size is therefore

$$A = \frac{(1.75)(27.168 \times 10^6 \text{ Btu})}{(1402 \frac{\text{Btu}}{\text{ft}^2\text{-day}})(31 \text{ days})}$$

$$= 1094 \text{ ft}^2$$

The curves in Figures 4-5 and 4-6 were developed for the case of collector slope equal to the local latitude, a house heating requirement of 15,000 Btu/DD and a daily hot water usage of 80 gallons/day. Tests were performed to determine the reliability of the curves for design conditions which deviate from those that were assumed. The house heating requirement was varied between 10,000 and 30,000 Btu/DD and the collector slope was varied between local latitude-15° and local latitude+15° for the case of space and service hot water heating. Very little change between respective standard deviations for the test cases and those given in Table 4-2 occurred. Therefore, the curves may be considered to be useful for design conditions between 10,000 and 30,000 Btu/DD and collector slopes between local latitude-15° and local latitude+15°.

A similar test was also performed for the case of service hot water heating only. The constraints of this test were that the hot water usage was between 50 and 200 gallons/day and the collector slope varied between latitude-15° and latitude+15°. Again, little change was found between the standard deviations obtained for this test and the standard deviations given in Table 4-2. Therefore, Figure 4-7 may be used to estimate collector sizing requirements for service hot water systems for any location, load, and collector tilt between latitude-15° and latitude+15°.

Separate design curves were found to be required for the case of collector slope equal to 90°. The same procedure discussed for the development of Figures 4-5 through 4-7 was again employed for the development of the design curves for this particular case. The resulting curves for the case of collector slope equal to 90° are given in Figures 4-8 through 4-10. The resulting standard deviations are given in Table 4-6.

Tests similar to those performed for the previous curves, i.e., tests to determine their usefulness for design conditions that differ from those assumed during the development of the curves, were also performed for the case of collector slope equal to 90°. Again, little change was seen between the standard deviations obtained during these tests and those outlined in Table 4-6. Therefore, Figures 4-8 and 4-9 are available for use when the house heating requirements vary between 10,000 and 30,000 Btu/DD for a collector tilted at 90°, and Figure 4-10 is available for use when the daily hot water usage varies between 50 and 200 gallons/day for a 90° collector slope.

It should be emphasized that this method should be used for preliminary design purposes only and that more accurate results would ordinarily be obtained by conducting a monthly analysis as described in Module 7. In addition, these results are limited to solar systems that are represented by the configurations shown in Figures 4-1 through 4-4.

Table 4-6. Standard Deviations for Design Curves of Figures 4-8 Through 4-10, Slope = 90°

Case Description	Figure	Coll. No.	Std. Dev.
Air System - Space Heating and Service Hot Water Heating	4-8	7	0.03331
Water System - Space Heating and Service Hot Water Heating	4-9	1	0.03138
		2	0.03093
		3	0.03318
		4	0.03147
		5	0.03168
		6	0.03164
Air and Water Systems - Service Hot Water Heating	4-10	1	0.04525
		2	0.04321
		3	0.04442
		4	0.04579
		5	0.03655
		6	0.03617
		7	0.03388

B. BALCOMB-HEDSTROM METHOD[2]

The objective of this method is to enable the designer to estimate the collector area required to supply an annual fraction of the load equal to 0.25, 0.50, or 0.75. The method requires monthly average radiation data and monthly average degree-day information.

The development of this procedure was based on a "standard" liquid collector and a "standard" air collector. The two "standard" systems used by the authors are presented in Tables 4-7 and 4-8. The results were found to be so close for each system that the estimation method was assumed to be the same for both system types.

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[2] J. D. Balcomb and J. C. Hedstrom, "A Simplified Method for Calculating Required Solar Collector Array Size for Space Heating," Sharing the Sun Solar Energy Conference, Winnipeg, Canada, August 1976.

Table 4-7. Standard Liquid-System Parameters for Balcomb-Hedstrom Procedure

Values of parameters used for the "standard" solar heating system using liquid-cooled solar collectors, a heat exchanger, water tank thermal storage, and a forced air heat distribution system to the building. The values are normalized to one square foot of collector (ft_c^2).

Solar Collectors		
Number of glazings	1	
Glass transmissivity (at normal incidence)	0.86	(6% absorption, 8% reflection)
Surface absorptance (solar)	0.98	
Surface emittance (IR)	0.89	
Back-insulation U-value	0.083	BTU/hr-°F-ft^2
Coolant flow rate	20	BTU/hr-°F-ft^2
Heat capacity	1	BTU/°F-ft_c^2
Heat transfer coefficient to liquid coolant	30	BTU/°F-ft_c^2
Tilt (from horizontal)	Latitude + 10 degrees	
Orientation	Due south	
Collector Plumbing		
Heat loss coefficient (to ambient)	0.04	BTU/hr-°F-ft_c^2
Heat Exchanger		
Heat transfer effectiveness	10	BTU/°F-hr-ft_c^2
Thermal Storage		
Heat capacity	15	BTU/°F-ft^2
Heat loss coefficient (i.e., assuming all heat loss is to heated space)	0	BTU/°F-hr-ft_c^2
Heat Distribution System		
Design air distribution temperature*	120°F	
Controls		
Building maintained at 68°F		
Collectors on when advantageous		

*The coil and air circulation are sized to meet the building load with an outside temperature of -2°F with 133°F water and airflow rate adequate to make up the space heat losses at an air discharge temperature of 120°F. This corresponds to a finned-tube coil effectiveness of 80 percent.

Table 4-8. Standard Air System Parameters for Balcomb-Hedstrom Procedure

Values of parameters used for the "standard" solar heating system using air heating collectors, a rock bed thermal storage, and a forced air heat distribution to the building. Values are normalized to one square foot of collector (ft$_c^2$).

Solar Collectors		
Number of glazings	1	
Glass transmissivity (at normal incidence)	0.86	(6% absorption, 8% reflection)
Glass absorptance	0.98	
Glass emittance	0.89	
Back-insulation U-value	0.083	BTU/hr-^{o}F-ft$_c^2$
Heat capacity	0.5	BTU/^{o}-ft$_c^2$
Airflow rate	2	CFM/ft$_c^2$
Heat transfer coefficient to air	4	BTU/hr-^{o}F-ft$_c^2$
Tilt	Latitude + 10 degrees	
Orientation	Due south	
Collector Air Ducts		
Heat loss coefficient (to ambient)	0.1	BTU/hr-^{o}F-ft$_c^2$
Thermal Storage		
Heat capacity	15	BTU/^{o}F-ft$_c^2$
Heat loss coefficient (i.e., assuming all heat loss is to heated space)	0	BTU/^{o}F-hr-ft$_c^2$
Dimensionless rock-bed heat transfer length*	10	
Heat Distribution System		
Airflow rate	2	CFM/ft$_c^2$
Controls		
Building maintained at 68^{o}F.		
Collectors on when advantageous		

*The rock-bed length (distance in the direction of flow) is greater than 5 times the relaxation length for heat transfer (15 was used in the model). Physically this means that the bed is at least 12 times as long as the rock diameter. It is important to note that the flow direction is reversed in the rock bed, being in one direction during the charging period and in the opposite direction during discharging.

A table of 85 locations throughout the United States and Canada listing three values for a parameter called LC was developed and is included here as Table 4-9. Each value of LC corresponds to an annual fraction of the load supplied by the solar system for each location. The corresponding f's are for values of 0.25, 0.50 and 0.75.

If the location in question is one of the 85 given locations, then the procedure to determine the required area for values of f = 0.25, 0.50 and 0.75 is an easy, two-step procedure. First, the user calculates the house heating requirement, QDES. The second step is to determine the required collector area corresponding to one of the three available values of f. This is evaluated by dividing the QDES obtained above by the value of LC for the particular f desired.

For example, if QDES = 15,000 Btu/DD and 75% of the required load is to be supplied by the solar system in Grand Junction, Colorado, then the required area is

$$QDES/LC_{0.75} = 15,000/22 = 680 \text{ square feet.}$$

To determine the required area for a location not given in Table 4-9, an iterative procedure is necessary. The procedure is begun by calculating QDES and selecting a site from the table of 85 locations climatically similar to the region under analysis. An approximate collector area can then be determined by the procedure above. The degree-days and solar insolation on a horizontal surface for each month must be determined for the new location. The solar insolation on the tilted surface is then calculated. Calculation of the 'solar load ratio,' SLR, is also required to refine the approximated area. The value of SLR is used to obtain the annual solar heating fraction. The result obtained for the annual solar

Table 4-9. Values of LC for 85 Selected Cities

City, State	Latitude °N	Elevation ft	Degree Days	LC, Btu/degree-day-ft^2_c where solar provides 25%, 50%, 75% of total heat 25%	50%	75%
Los Alamos, NM	36	7200	6600	107	41	21
Columbus, OH	40	760	5211	77	29	13
Corvallis, OR	45	236	4726	120	42	18
Davis, CA	39	50	2502	198	72	33
East Lansing, MI	43	878	6909	76	28	13
East Wareham, MA	42	50	5891	97	37	18
El Centro, CA	33	12	1458	547	206	97
Flaming Gorge, UT	41	6273	6929	111	43	21
Granby, CO	40	8340	5524	119	47	23
Toronto, Canada	44	443	6827	72	27	13
Griffin, GA	33	1001	2136	217	84	42
Winnipeg, Canada	50	820	10629	63	23	11
Ithaca, NY	42	951	6914	68	24	11
Inyokern, CA	36	2186	3528	232	88	42
ANL, Lemont, IL	42	750	6155	79	30	14
Newport, RI	41	50	5804	97	37	18
Laramie, WY	41	7240	7381	106	42	21
Page, AZ	37	4280	6632	128	48	23
Prosser, WA	46	840	4805	117	41	18
Pullman, WA	47	2583	5542	100	36	16
Put-In-Bay, OH	42	580	5796	68	24	11
Richland, WA	47	731	5941	100	35	15
Raleigh, NC	36	440	3393	133	52	25
Riverside, CA	34	1050	1803	391	152	74
Seattle, WA	48	110	4785	94	33	13
Sayville, NY	41	56	4811	98	38	18
Schenectady, NY	43	490	6650	63	24	11
Seabrook, NY	39	110	4812	97	37	18
Shreveport, LA	32	220	2184	179	70	35
State College, PA	41	1230	5934	78	29	14
Stillwater, OK	36	910	3725	132	52	25
Tallahassee, FL	30	64	1485	283	113	57
Tucson, AZ	32	2440	1800	301	118	59
Oak Ridge, TN	36	940	3817	111	42	20
Fort Worth, TX	33	574	2405	186	73	37
Lake Charles, LA	30	60	1459	244	96	48
Apalachicola, FL	30	46	1308	324	129	65
Brownsville, TX	26	48	600	517	218	110
San Antonio, TX	30	818	1546	262	103	52
Greensboro, NC	36	914	3805	128	50	24
Hatteras, NC	35	27	2612	204	79	39
Atlanta, GA	34	1018	2961	154	59	29
Charleston, SC	33	69	2033	210	82	41

Table 4-9. (continued)

City, State	Latitude °N	Elevation ft	Degree Days	LC, Btu/degree-day-ft_c^2 where solar provides 25%, 50%, 75% of total heat		
				25%	50%	75%
Nashville, TN	36	614	3578	117	44	21
Lake Charles, LA	30	39	1459	261	104	53
Little Rock, AR	35	276	3219	126	48	24
Oklahoma City, OK	35	1317	3725	134	53	26
Columbia, MO	39	814	5046	102	38	18
Dodge City, KA	38	2625	4986	126	49	24
Caribou, ME	47	640	9767	68	26	12
Burlington, VT	44	385	8269	63	24	11
Blue Hill, MA	42	670	6368	82	31	15
Cleveland, OH	41	871	6351	71	26	12
Madison, WI	43	889	7863	76	28	13
Sault Ste. Marie, MI	46	724	9048	74	27	12
Saint Cloud, MN	46	1062	8879	71	27	13
Lincoln, NE	41	1316	5864	104	39	19
Midland, TX	32	2885	2591	202	79	39
El Paso, TX	32	3954	2700	228	88	44
Albuquerque, NM	35	5327	4348	161	64	31
Grand Junction, CO	39	4832	5641	119	46	22
Ely, NV	39	6279	7733	119	47	23
Las Vegas, NV	36	2188	2709	218	84	42
Phoenix, AZ	33	1139	1765	300	118	59
Reno, NV	39	4400	6832	125	47	22
Santa Maria, CA	35	289	2967	353	142	67
Bismarck, ND	47	1677	8851	78	29	14
Lander, WY	43	5574	7870	108	42	21
Glasgow, MT	48	2109	2996	105	41	20
Rapid City, SD	44	3180	7345	97	37	18
Salt Lake City, UT	41	4238	6052	107	40	19
Boise, ID	44	2895	5809	108	39	17
Great Falls, MT	47	3692	7750	93	35	16
Spokane, WA	48	2356	6655	90	31	14
Medford, OR	42	1321	5008	107	38	16
Los Angeles, CA	34	540	2061	416	157	75
Fresno, CA	37	336	2492	195	70	32
Silver Hill, MD	39	292	4224	111	43	21
Cape Hatteras, NC	35	27	4612	189	74	36
Sterling, VA	39	276	4224	111	43	21
Indianapolis, IN	40	819	5699	86	32	15
Astoria, OR	46	22	5186	127	45	19
Boston, MA	42	157	5624	86	33	16
New York, NY	41	187	4871	88	34	16
North Omaha, NE	41	1323	6612	89	34	16

heating fraction should then confirm the fraction of annual heat load assumed initially. If the agreement is not satisfactory, then the collector area should be adjusted and the last two steps repeated.

An example follows for Terre Haute, Indiana, which is not listed among the 85 given locations. A "standard" system has been assumed and is expected to supply 75% of the required load. What collector area should be used? Assume QDES = 15,000 Btu/DD.

EXAMPLE 1

Step 1 QDES assumed to be 15,000 Btu/DD

Step 2 Using Indianapolis, Indiana, as a location climatically similar to that of Terre Haute, the estimated area is QDES/LC = 15,000/15 = 1000 square feet

Step 3 Degree-days and solar radiation on a horizontal surface for each month (Btu/FT^2)

Table 4-10. Monthly Degree-Days and Solar Radiation on a Horizontal Surface for Indianapolis, Indiana

MONTH	DD	$\overline{H}$	MONTH	DD	$\overline{H}$
Jan	1113	16,312	Jul	0	63,225
Feb	949	22,327	Aug	0	56,795
Mar	809	36,707	Sep	90	45,399
Apr	432	44,436	Oct	316	33,926
May	177	56,668	Nov	723	19,872
Jun	39	61,260	Dec	1051	15,224

Step 4 Solar radiation on the tilted collector (assumed by Balcomb and Hedstrom to be tilted at latitude $+10^0$)

$$\begin{bmatrix}\text{Total radiation on} \\ \text{Collector Surface}\end{bmatrix} \approx 1.025\ Y - 8200$$

$$\text{where } Y = \frac{\text{total monthly radiation on horizontal surface}}{\cos\ (\text{latitude} - \text{solar declination (at midmonth)})}$$

and where

$$\begin{bmatrix}\text{Solar declination} \\ \text{at midmonth}\end{bmatrix} \approx 23.45 \cos\ (30M - 187)$$

and M = month number, i.e., January = 1, December = 12

Table 4-11. Monthly Solar Insolation on a Tilted Surface for Indianapolis, Indiana, by the Balcomb-Hedstrom Procedure

MONTH	$\overline{H}_T$	MONTH	$\overline{H}_T$
Jan	26,104	Jul	60,090
Feb	29,548	Aug	56,726
Mar	42,549	Sep	50,788
Apr	44,366	Oct	45,564
May	53,893	Nov	31,475
Jun	52,040	Dec	26,290

Step 5 Determine Solar Load Ratio (SLR) for each month

$$\text{SLR}_i = \frac{[\text{collector area}] \times [\overline{H}_T]_i}{\text{QDES} \times \text{DD}_i},\ i = 1, 12$$

Table 4-12. Monthly Solar Load Ratios for Boulder, Colorado

MONTH	SLR	MONTH	SLR
Jan	1.56	Jul	--
Feb	2.08	Aug	--
Mar	3.51	Sep	37.62
Apr	6.85	Oct	9.61
May	20.30	Nov	2.90
Jun	88.96	Dec	1.67

Step 6 Determine the Annual Solar Heating Fraction (ASHF)

$$\text{ASHF} = \frac{\sum\limits_{\text{month}=1}^{12} (\text{DD})(\text{X})}{\sum\limits_{\text{month}=1}^{12} (\text{DD})}$$

where

$X = 1.06 - 1.366 \exp(-0.55 \text{ SLR}) + 0.306 \exp(-1.05 \text{ SLR})$

(FOR SLR $<$ 5.66) and

$X = 1.0$

(FOR SLR $\geq$ 5.66)

Table 4-13. Monthly X-Factor for Determining Annual Solar Heating Fraction for Balcomb-Hedstrom Procedure

MONTH	X	MONTH	X
Jan	0.54	Jul	1.00
Feb	0.66	Aug	1.00
Mar	0.87	Sep	1.00
Apr	1.00	Oct	1.00
May	1.00	Nov	0.80
Jun	1.00	Dec	0.57

Then from the above equation we obtain

ASHF = 4162/5699 = 0.73

The resulting annual solar heating fraction, therefore, indicates that a larger area should now be tested so that the annual fraction of the load supplied by solar is closer to 0.75. The test was not carried out any further. The procedure outlined in Steps 1 through 6 required approximately one hour.

DESIGN GUIDELINES

We have seen that the primary information required for a quick, approximate design of a solar system is the mean daily solar insolation and the average ambient temperature or the average degree days for the given location. By knowing the degree days for the location, or the average ambient temperature, and by knowing the design heat load for the structure, the average heat load may be readily determined. The percentage of this heat load that may be supplied by a solar system is approximated by knowing the mean daily solar insolation on a horizontal surface at the selected location. The mean solar insolation on the horizontal surface is converted to mean radiation on a tilted surface by means of approximate rules or charts. The collector may then be sized by one of the methods just presented.

Once the size of the collector array has been determined, the storage volume may be established by assuming one-half cubic foot of rocks per square foot of collector for air systems with pebble-bed storage or by assuming 1.5 gallons of water per square foot of collector for water systems. The flow rates of transport media through the collectors may be assumed to be 2 CFM per square foot of collector for air systems and about 0.025 gallons per minute per square foot of collector for water systems. Additional components such as pumps, blowers, and heat exchangers may then be sized from the above flow rate information. These components will be discussed in more detail in Module 16.

Economic analyses are conducted after the preliminary sizing studies are performed. The system costs may be estimated on the basis of the preliminary sizing calculations. These system costs are then used in economic analyses in order to provide the client with enough information that decisions regarding that design may be made. There are available a number of algorithms for performing economic investigations. The details are discussed in Module 8.

The design procedure described above is illustrated in Figures 4-11 and 4-12, and is outlined below in more specific terms.

STEP BY STEP DESIGN PROCEDURE

1. Determine the design heat load for the building in Btu/DD. (British Thermal Units per degree day).

2. Determine the number of heating degree days for the given location.

3. Multiply the heat load determined by Step 1 by the degree days determined in Step 2 to determine the heating load for the building.

4. Determine the monthly average solar radiation on a horizontal surface, $\overline{H}$, at the given location.

5. Convert the monthly average radiation on a horizontal surface to monthly average radiation on the desired tilted surface to obtain $\overline{H}_T$.

6. Estimate the required collector area by one of the methods presented above.

7. Determine storage size (gallons) for water systems by multiplying the collector size (ft^2) by 1.5 gallons/ft^2. For air systems using pebble storage multiply the collector size by 0.5 ft^3 rocks/ft^2 of collector.

8. Size pumps or blowers from the requirements that, for water systems, the collector flow rate should be about 0.025 gallons per minute

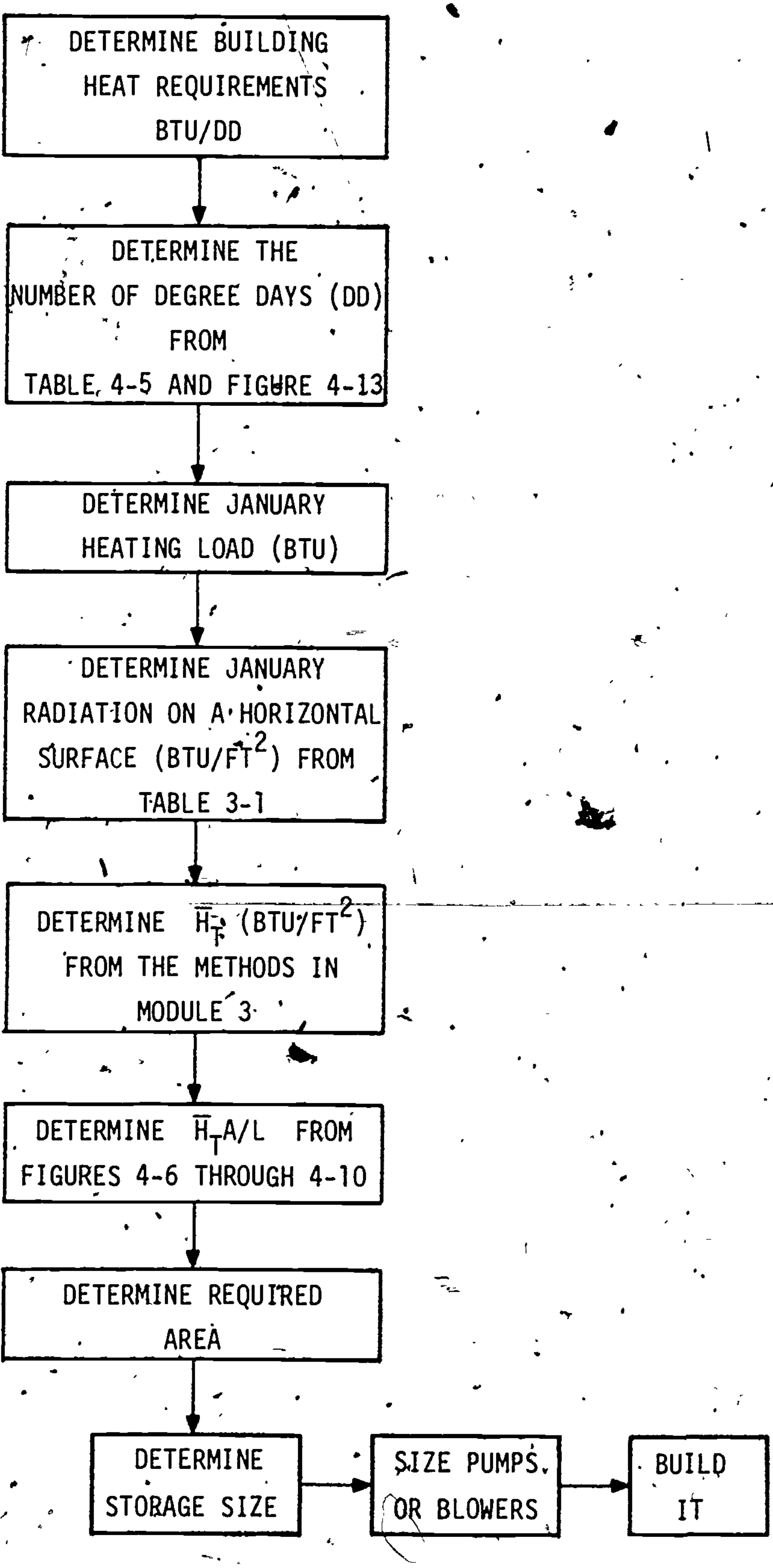

DETERMINE BUILDING HEAT REQUIREMENTS BTU/DD
DETERMINE THE NUMBER OF DEGREE DAYS (DD) FROM TABLE 4-5 AND FIGURE 4-13
DETERMINE JANUARY HEATING LOAD (BTU)
DETERMINE JANUARY RADIATION ON A HORIZONTAL SURFACE (BTU/FT^2) FROM TABLE 3-1
DETERMINE $\overline{H}_T$ (BTU/FT^2) FROM THE METHODS IN MODULE 3
DETERMINE $\overline{H}_TA/L$ FROM FIGURES 4-6 THROUGH 4-10
DETERMINE REQUIRED AREA
DETERMINE STORAGE SIZE
SIZE PUMPS OR BLOWERS
BUILD IT

Figure 4-11. Huck-Winn Design Procedure

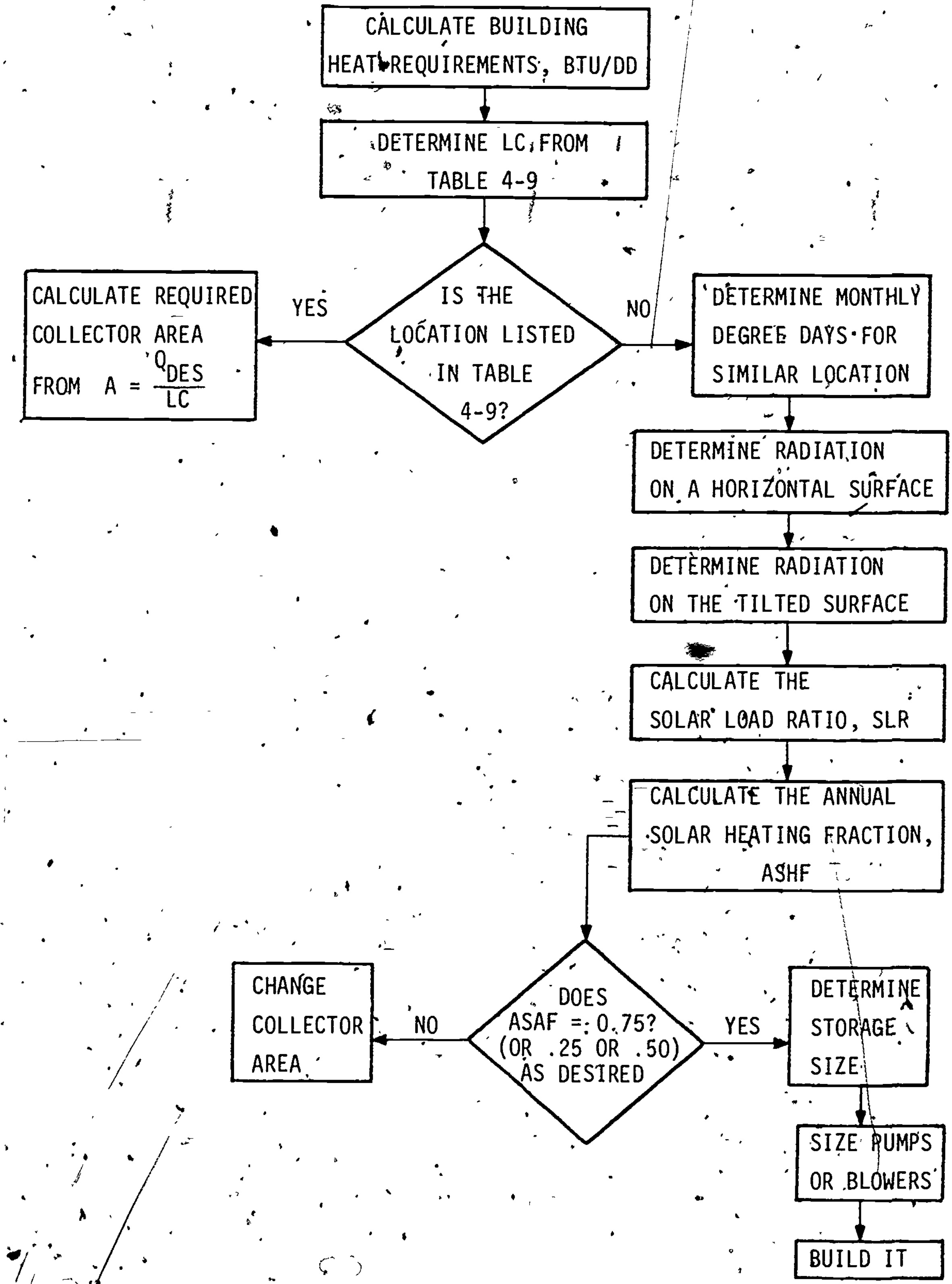

CALCULATE BUILDING HEAT REQUIREMENTS, BTU/DD
DETERMINE LC FROM TABLE 4-9
IS THE LOCATION LISTED IN TABLE 4-9?
YES
NO
CALCULATE REQUIRED COLLECTOR AREA FROM $A = \frac{Q_{DES}}{LC}$
DETERMINE MONTHLY DEGREE DAYS FOR SIMILAR LOCATION
DETERMINE RADIATION ON A HORIZONTAL SURFACE
DETERMINE RADIATION ON THE TILTED SURFACE
CALCULATE THE SOLAR LOAD RATIO, SLR
CALCULATE THE ANNUAL SOLAR HEATING FRACTION, ASHF
DOES ASAF = 0.75? (OR .25 OR .50) AS DESIRED
NO
CHANGE COLLECTOR AREA
YES
DETERMINE STORAGE SIZE
SIZE PUMPS OR BLOWERS
BUILD IT

Figure 4-12. Balcomb-Hedstrom Design Procedure

per square foot of collector, and for air systems, the collector flow rate should be about 2 CFM per square foot of collector.

DESIGN DATA

TEMPERATURE DATA

Figures 4-13 through 4-24 present degree-day information for the United States. These figures represent the average total heating degree days by month for the entire year. These were taken from the Climatic Atlas for the United States published by the U.S. Department of Commerce. By knowing the design heat load for a building, one can approximate the total heat load for each month of the year from these contour maps. For example, consider a house to be located in Boulder, Colorado. Suppose the design heat load is determined to be 57,000 Btu/hr. First, we must convert this to Btu per degree day. This is accomplished by the following calculation. The design temperature for Boulder is 8^0F, as determined from the ASHRAE Handbook of Fundamentals. Therefore, the house heat load in Btu/DD is

$$Q\left(\frac{Btu}{DD}\right) = \frac{(57{,}000\ Btu/hr)\cdot(24\ hr/day)}{57\ DD/day}$$

$$= 24{,}000\ Btu/DD$$

From the contour map for January (Figure 4-13) one can see that Boulder has approximately 1200 degree days for the month of January. Consequently, the January heat load would be 24,000 Btu per degree day times 1200 degree days or 28.8 million Btu for the month of January.

Table 4-5 presents monthly and total annual degree-day information for selected locations throughout the United States. This table may be used in place of the contour maps when the desired location is included in the table.

SOLAR DATA

The solar insolation data were presented in Module 3. Figures 3-1 through 3-12 present the mean daily solar radiation on a horizontal surface for each month for the United States. From Figure 3-1, for example, we see that the January insolation for Boulder (40^0 N latitude) is approximately 200 Langleys or 738 Btu per square foot per day. This information must be converted to the radiation on a tilted surface in order to estimate the amount of heat that a collector array will collect at a given tilt angle. This is accomplished by referring to Tables 3-5 and 3-6 which give the conversion factors for determining radiation on tilted surfaces. $\overline{K}_T$ for Boulder is approximately 0.55. By interpolating between these two tables, assuming the tilt is equal to the latitude, we obtain $\overline{R} = 1.78$. Therefore the average daily radiation on the tilted surface for January in Boulder is 1314 Btu/ft^2.

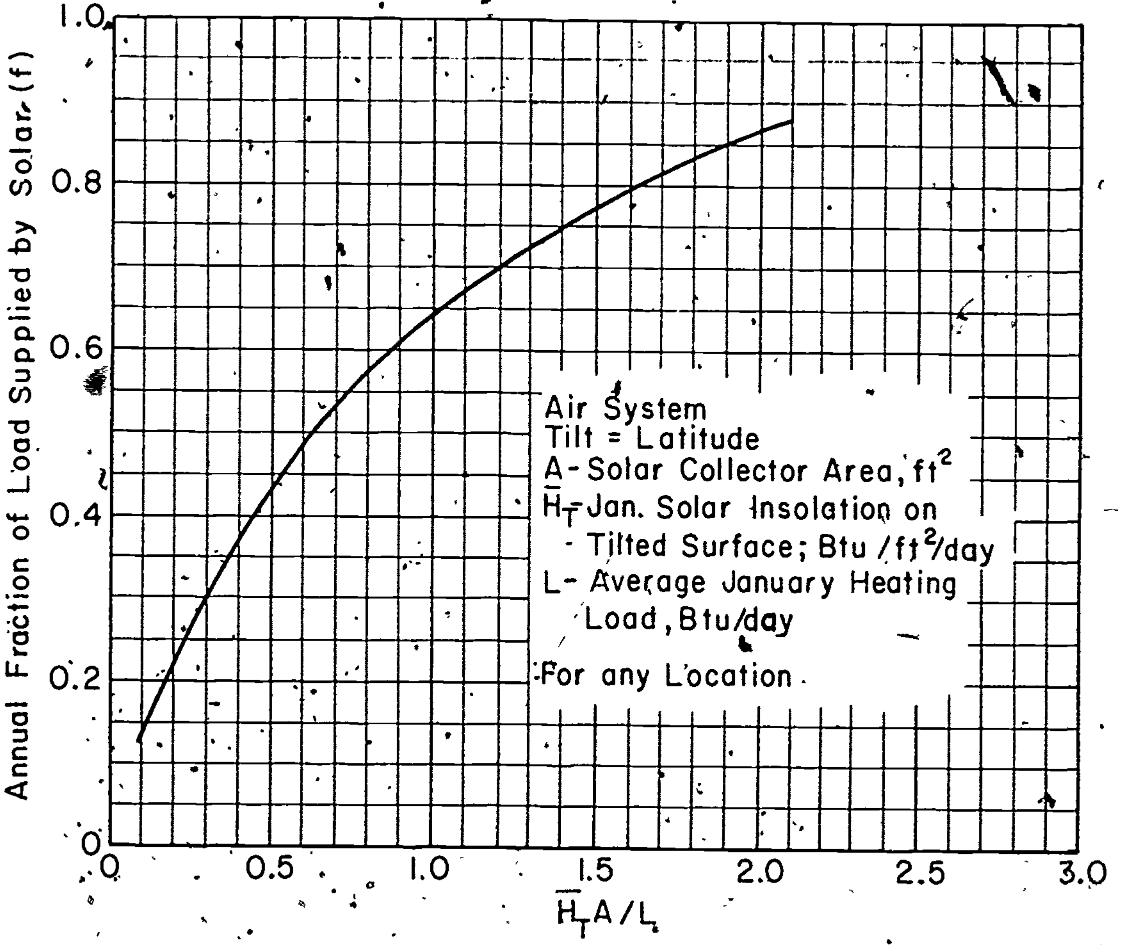


Figure 4-5. Fraction of Annual Load Supplied by Solar as a Function of $\overline{H}_T A/L$ for General Locations - Air System

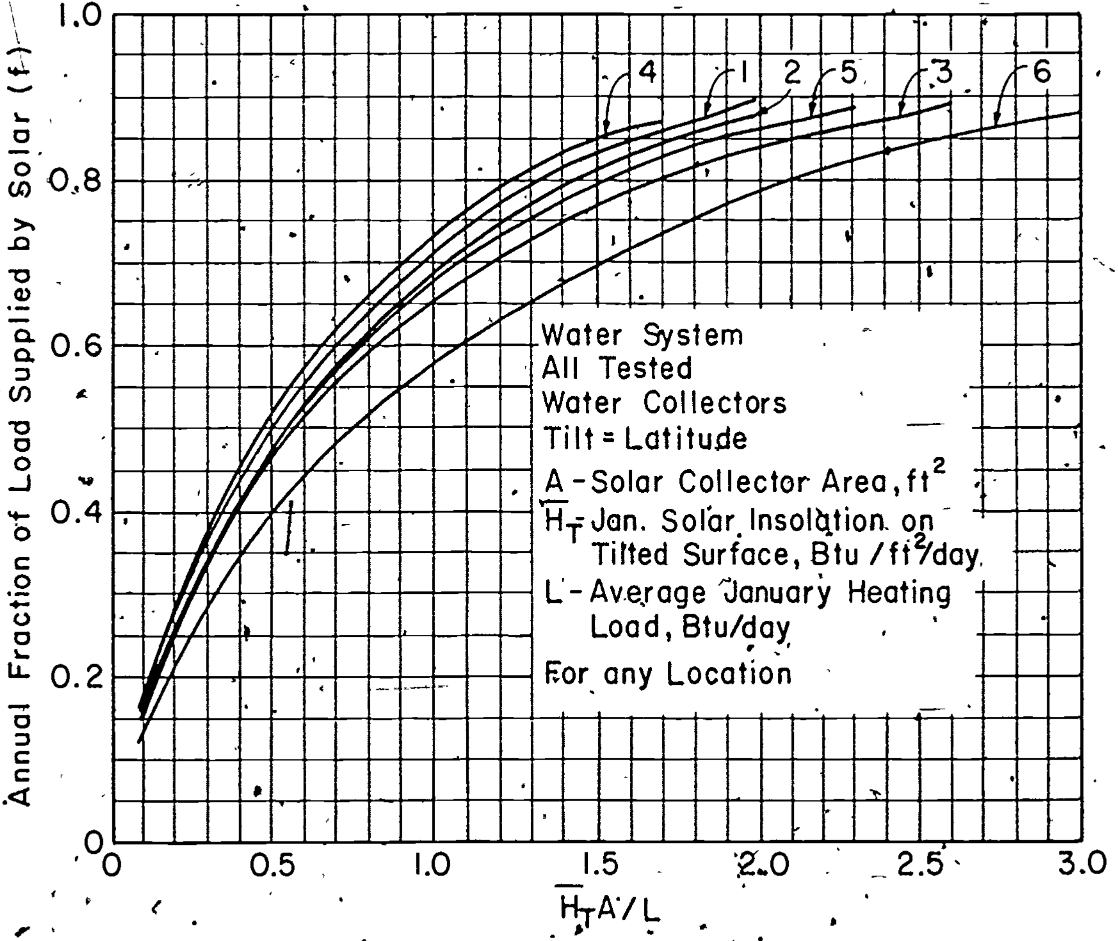


Figure 4-6. Fraction of Annual Load Supplied by Solar as a Function of $\overline{H}_T$ A/L for the Six Water Collectors Tested for General Locations

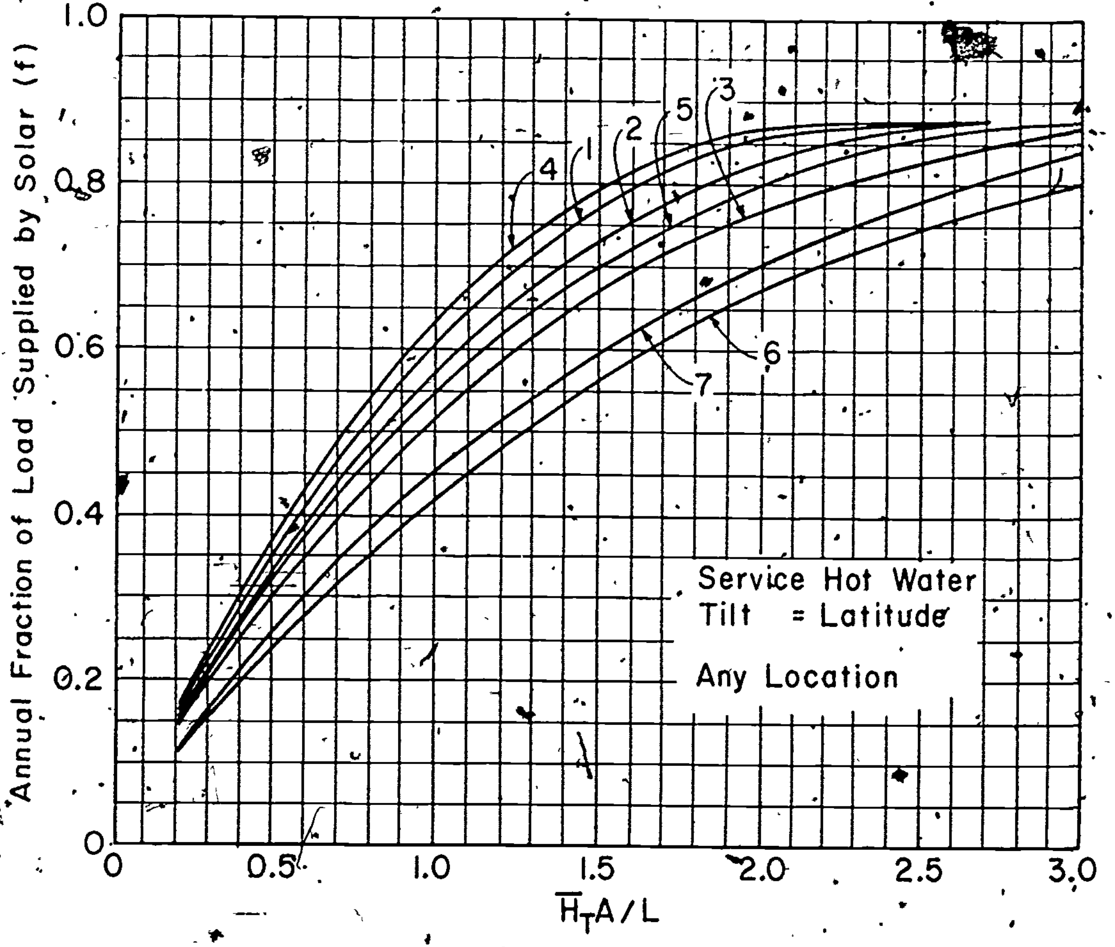


Figure 4-7. Fraction of Annual Load Supplied by Solar as a Function of $\overline{H}_T A/L$ for All Tested Collectors and for General Locations

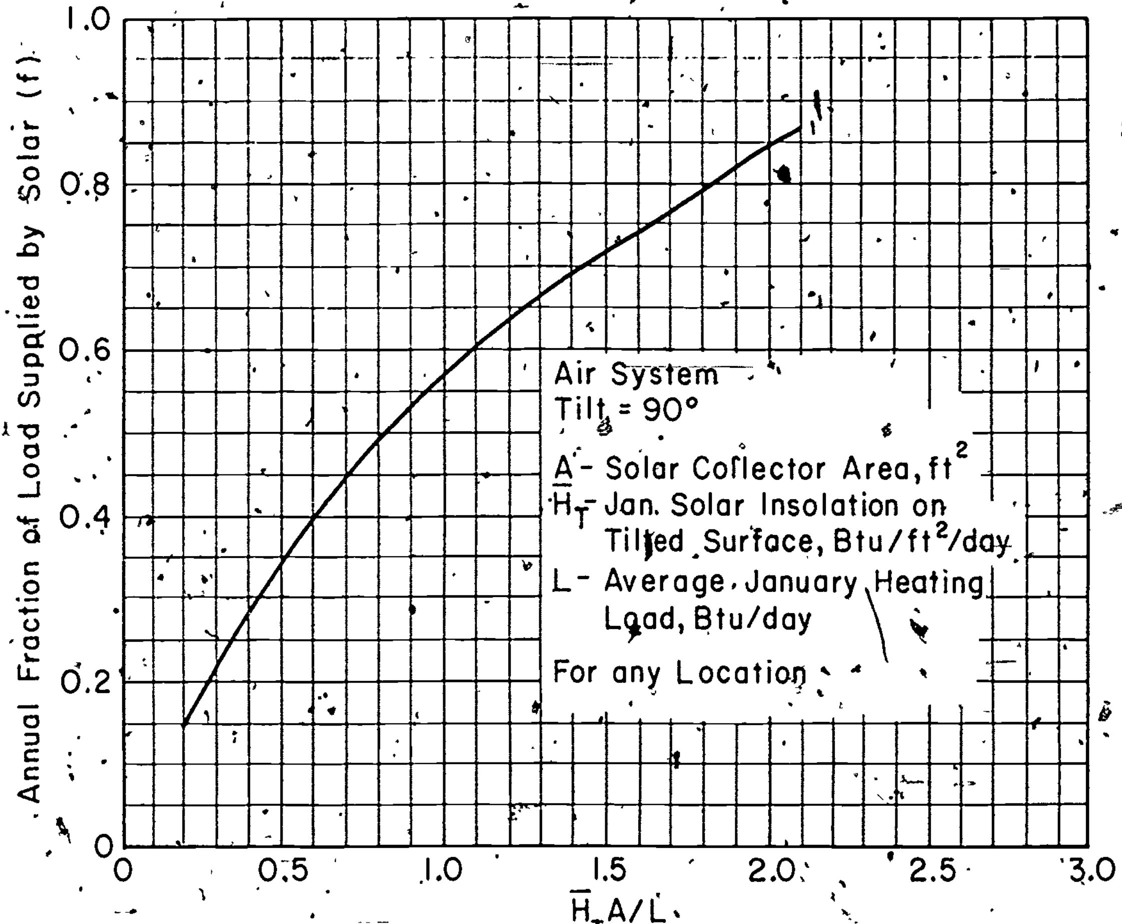


Figure 4-8. Fraction of Annual Load Supplied by Solar as a Function of $\bar{H}_T A/L$ for Any Location, Slope = 90° - Air System

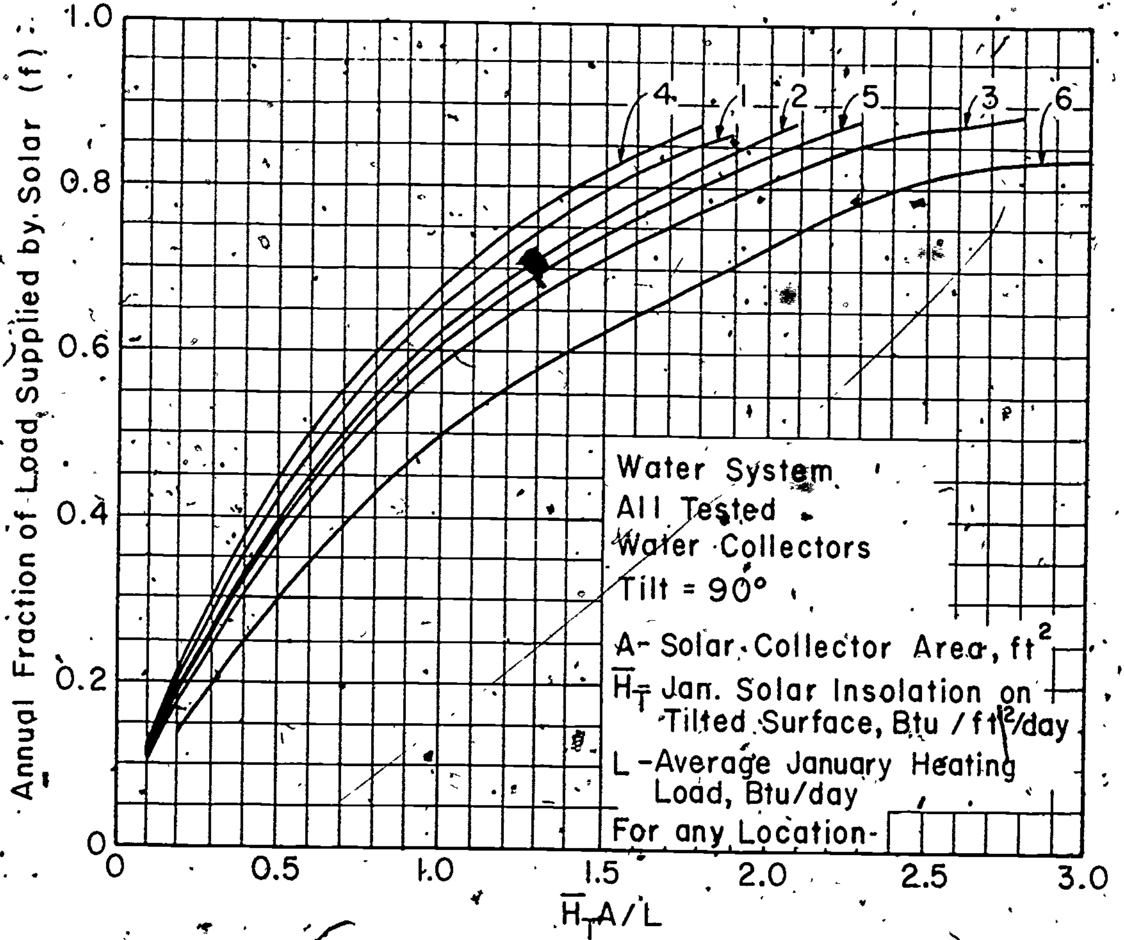


Figure 4-9. Fraction of Annual Load Supplied by Solar as a Function of $\bar{H}_T A/L$ for Any Location, Slope = 90^0 - Water System

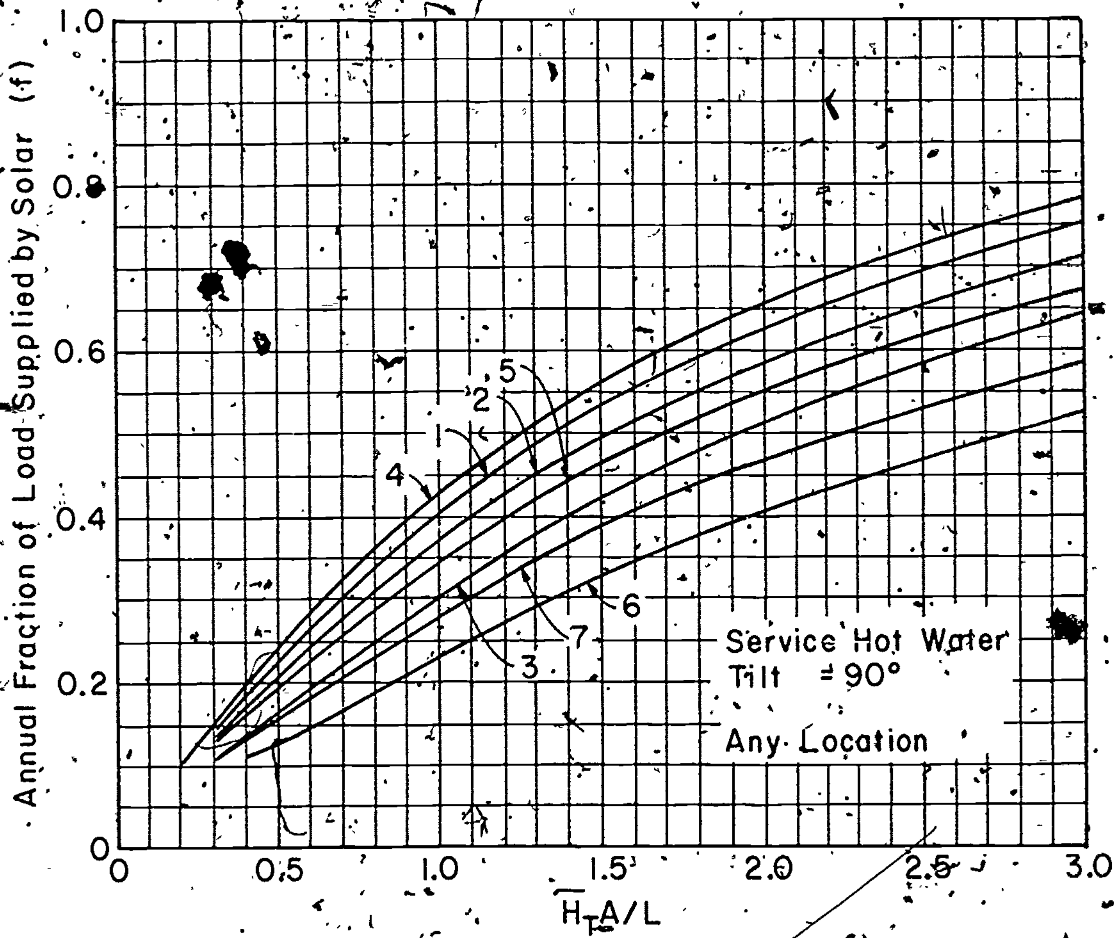
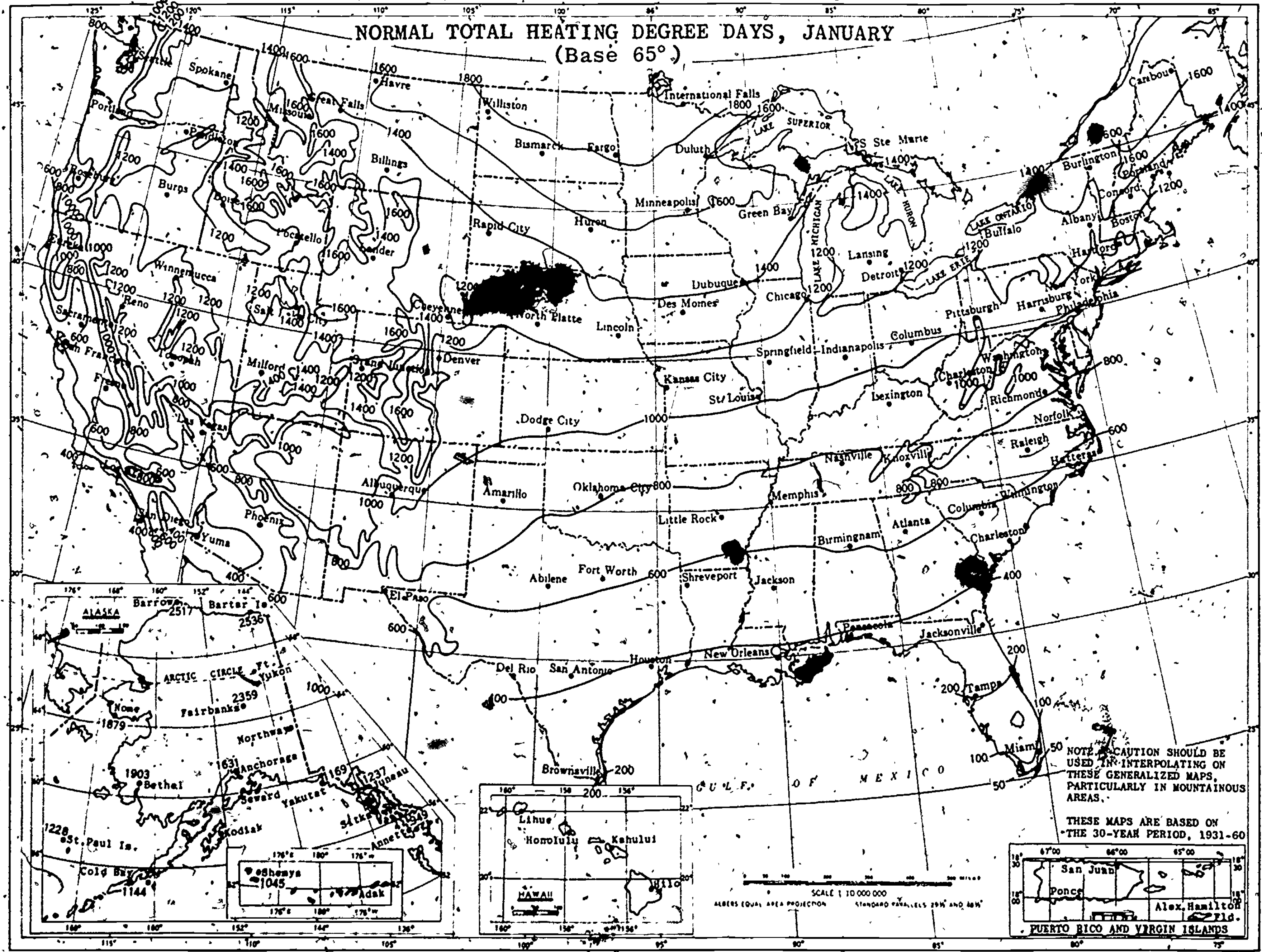


Figure 4-10. Fraction of Annual Load Supplied by Solar as a Function of $\overline{H}_T A/L$ for All Tested Collectors for Slope = 90° for General Locations

Figure 4-13



NORMAL TOTAL HEATING DEGREE DAYS, JANUARY
(Base 65°)
Seattle
Spokane
Portland
Pendleton
Roseburg
Burns
Boise
Eureka
Winnemucca
Reno
Sacramento
San Francisco
Fresno
Tonopah
Las Vegas
Los Angeles
San Diego
Yuma
Phoenix
Havre
Great Falls
Missoula
Billings
Pocatello
Lander
Salt Lake City
Milford
Grand Junction
Cheyenne
Denver
Albuquerque
El Paso
Williston
Bismarck
Fargo
Rapid City
Huron
North Platte
Lincoln
Dodge City
Amarillo
Oklahoma City
Abilene
Fort Worth
Del Rio
San Antonio
Houston
Brownsville
International Falls
Duluth
Minneapolis
Green Bay
Dubuque
Des Moines
Chicago
Kansas City
St Louis
Springfield
Indianapolis
Little Rock
Shreveport
Memphis
Jackson
New Orleans
S Ste Marie
Lansing
Detroit
Columbus
Lexington
Nashville
Knoxville
Birmingham
Atlanta
Pensacola
Jacksonville
Tampa
Miami
Pittsburgh
Harrisburg
Philadelphia
New York
Washington
Charleston
Richmond
Norfolk
Raleigh
Hatteras
Wilmington
Columbia
Charleston
Buffalo
Albany
Hartford
Boston
Concord
Burlington
Portland
Caribou
LAKE SUPERIOR
LAKE MICHIGAN
LAKE HURON
LAKE ONTARIO
LAKE ERIE
GULF OF MEXICO
ALASKA
Barrow
2517
Barter Is.
2536
ARCTIC CIRCLE
Ft. Yukon
Nome
1879
Fairbanks
2359
Northway
Bethel
1903
Anchorage
1631
Seward
Yakutat
1169
Juneau
1237
Sitka
Annette
Kodiak
St. Paul Is.
1228
Cold Bay
1144
Shemya
1045
Adak
HAWAII
Lihue
Honolulu
Kahului
Hilo
PUERTO RICO AND VIRGIN ISLANDS
San Juan
Ponce
Alex.Hamilton Fld.
SCALE 1:10 000 000
ALBERS EQUAL AREA PROJECTION
STANDARD PARALLELS 29½° AND 45½°

NOTE: CAUTION SHOULD BE USED IN INTERPOLATING ON THESE GENERALIZED MAPS, PARTICULARLY IN MOUNTAINOUS AREAS.

THESE MAPS ARE BASED ON THE 30-YEAR PERIOD, 1931-60

Figure 4-14

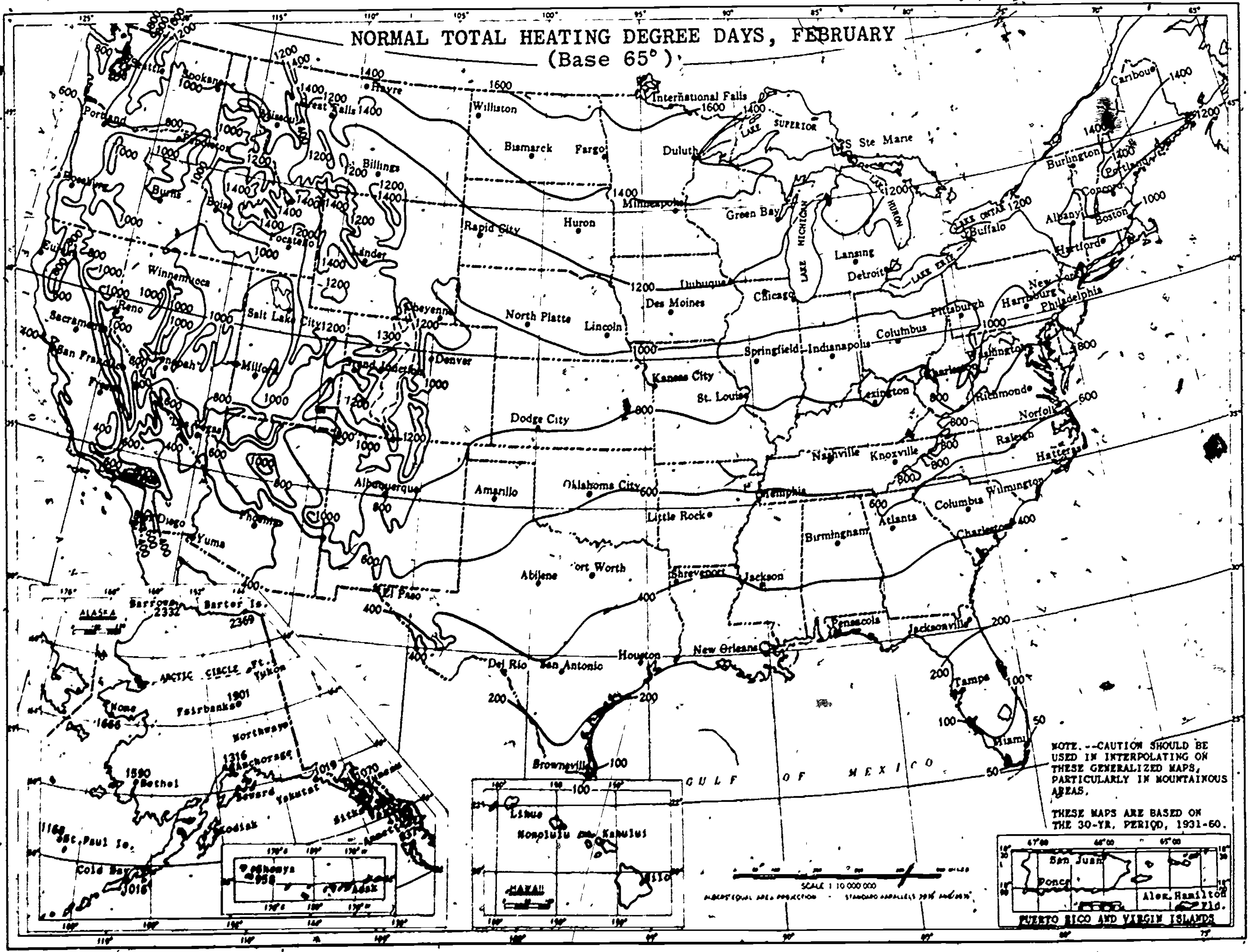
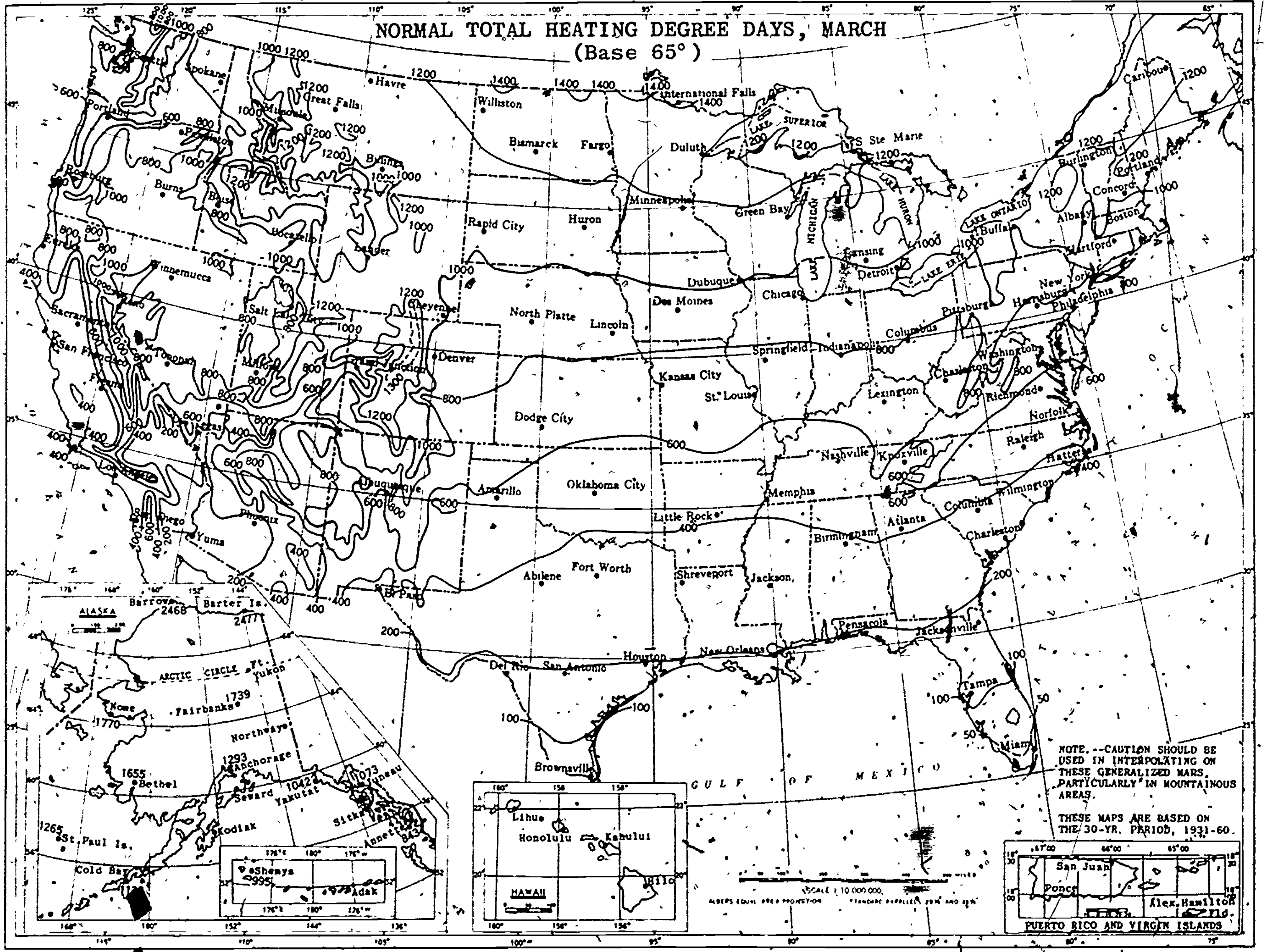

NORMAL TOTAL HEATING DEGREE DAYS, FEBRUARY
(Base 65°)
Seattle
Spokane
Portland
Pendleton
Roseburg
Burns
Boise
Eureka
Winnemucca
Reno
Sacramento
San Francisco
Fresno
Tonopah
Las Vegas
San Diego
Yuma
Phoenix
Salt Lake City
Milford
Pocatello
Lander
Great Falls
Missoula
Havre
Billings
Cheyenne
Grand Junction
Denver
Albuquerque
El Paso
Williston
Bismarck
Fargo
Rapid City
Huron
North Platte
Lincoln
Dodge City
Amarillo
Oklahoma City
Abilene
Fort Worth
Del Rio
San Antonio
Houston
Brownsville
International Falls
Duluth
Minneapolis
Des Moines
Dubuque
Kansas City
St. Louis
Little Rock
Shreveport
Jackson
New Orleans
Memphis
Nashville
Birmingham
Atlanta
Pensacola
Jacksonville
Tampa
Miami
Green Bay
Chicago
Springfield
Indianapolis
Lansing
Detroit
Columbus
Lexington
Knoxville
S Ste Marie
Lake Superior
Lake Michigan
Lake Huron
Lake Ontario
Lake Erie
Buffalo
Pittsburgh
Charleston
Harrisburg
Philadelphia
New York
Washington
Richmond
Norfolk
Raleigh
Hatteras
Wilmington
Columbia
Charleston
Albany
Hartford
Boston
Concord
Burlington
Portland
Caribou
Gulf of Mexico
1600
1400
1200
1000
800
600
400
200
100
50
ALASKA
Barrow
2332
Barter Is.
2369
Arctic Circle
Ft. Yukon
Fairbanks
1901
Nome
1666
Northway
Anchorage
1316
Bethel
1590
Seward
Yakutat
1019
Juneau
1070
Sitka
Annette
831
Kodiak
St. Paul Is.
1164
Cold Bay
1016
Shemya
958
Adak
HAWAII
Lihue
Honolulu
Kahului
Hilo
100
SCALE 1:10 000 000
ALBERS EQUAL AREA PROJECTION - STANDARD PARALLELS 29½° AND 45½°
NOTE.--CAUTION SHOULD BE USED IN INTERPOLATING ON THESE GENERALIZED MAPS, PARTICULARLY IN MOUNTAINOUS AREAS.
THESE MAPS ARE BASED ON THE 30-YR. PERIOD, 1931-60.
San Juan
Ponce
Alex. Hamilton Fld.
PUERTO RICO AND VIRGIN ISLANDS

Figure 4-15

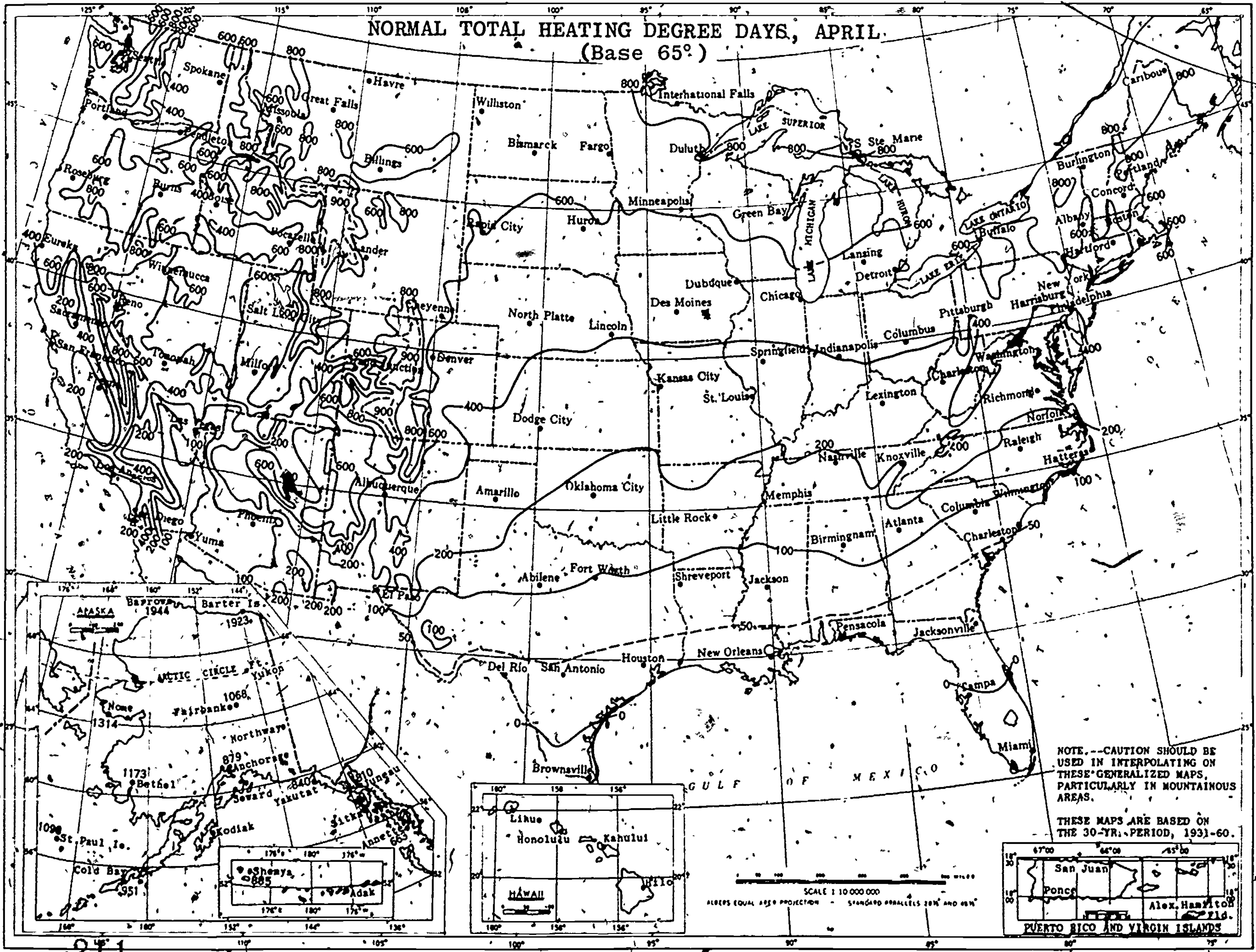

NORMAL TOTAL HEATING DEGREE DAYS, MARCH
(Base 65°)

Seattle
Spokane
Portland
Roseburg
Burns
Boise
Eureka
Winnemucca
Reno
Sacramento
San Francisco
Tonopah
Fresno
Los Angeles
San Diego
Yuma
Las Vegas
Phoenix
Salt Lake City
Milford
Pocatello
Great Falls
Missoula
Havre
Billings
Lander
Cheyenne
Grand Junction
Denver
Albuquerque
El Paso
Williston
Bismarck
Fargo
Rapid City
Huron
North Platte
Lincoln
Dodge City
Amarillo
Oklahoma City
Abilene
Fort Worth
Del Rio
San Antonio
Houston
Brownsville
International Falls
Duluth
Minneapolis
Green Bay
Des Moines
Dubuque
Kansas City
St. Louis
Little Rock
Shreveport
Jackson
New Orleans
Memphis
Nashville
Knoxville
Birmingham
Atlanta
Pensacola
Jacksonville
Tampa
Miami
Chicago
Lansing
Detroit
Springfield
Indianapolis
Columbus
Lexington
Charleston
Pittsburgh
Harrisburg
Philadelphia
New York
Washington
Richmond
Norfolk
Raleigh
Hatteras
Wilmington
Columbia
Charleston
S Ste Marie
Buffalo
Albany
Hartford
Boston
Concord
Portland
Burlington
Caribou
LAKE SUPERIOR
LAKE MICHIGAN
LAKE HURON
LAKE ERIE
LAKE ONTARIO
GULF OF MEXICO
ALASKA
Barrow 2468
Barter Is. 2477
ARCTIC CIRCLE
Ft. Yukon
Nome 1770
Fairbanks 1739
Northway
Anchorage 1293
Bethel 1655
Seward
Yakutat 1042
Juneau 1073
Sitka
Annette 843
Kodiak
St. Paul Is. 1265
Cold Bay
Shemya 995
Adak
HAWAII
Lihue
Honolulu
Kahului
Hilo
PUERTO RICO AND VIRGIN ISLANDS
San Juan
Ponce
Alex. Hamilton Fld.
1400
1200
1000
800
600
400
200
100
50
SCALE 1:10,000,000
ALBERS EQUAL AREA PROJECTION STANDARD PARALLELS 29½° AND 45½°

NOTE.--CAUTION SHOULD BE USED IN INTERPOLATING ON THESE GENERALIZED MAPS, PARTICULARLY IN MOUNTAINOUS AREAS.

THESE MAPS ARE BASED ON THE 30-YR. PERIOD, 1931-60.

Figure 4-16



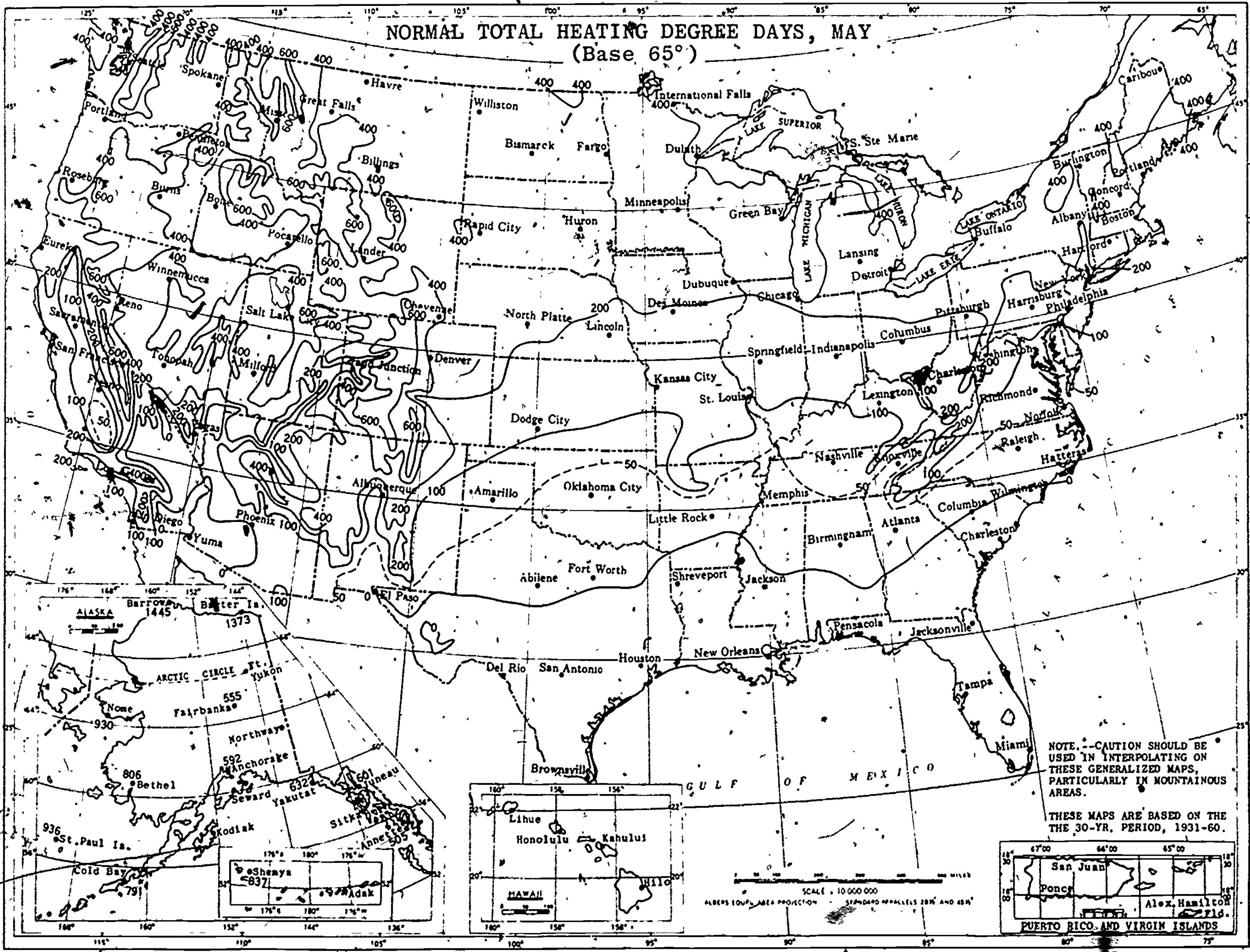
NORMAL TOTAL HEATING DEGREE DAYS, APRIL
(Base 65°)

NOTE.--CAUTION SHOULD BE USED IN INTERPOLATING ON THESE GENERALIZED MAPS, PARTICULARLY IN MOUNTAINOUS AREAS.

THESE MAPS ARE BASED ON THE 30-YR. PERIOD, 1931-60.

Seattle
Spokane
Portland
Pendleton
Roseburg
Burns
Boise
Eureka
Winnemucca
Reno
Sacramento
San Francisco
Fresno
Tonopah
Milford
Las Vegas
Los Angeles
San Diego
Yuma
Phoenix
Albuquerque
El Paso
Havre
Great Falls
Missoula
Billings
Pocatello
Lander
Salt Lake City
Cheyenne
Grand Junction
Denver
Williston
Bismarck
Fargo
Rapid City
Huron
North Platte
Lincoln
Dodge City
Amarillo
Oklahoma City
Abilene
Fort Worth
Del Rio
San Antonio
Houston
Brownsville
International Falls
Duluth
Minneapolis
Green Bay
Dubuque
Des Moines
Chicago
Kansas City
St. Louis
Springfield
Indianapolis
Little Rock
Shreveport
Jackson
Memphis
Nashville
New Orleans
Lansing
Detroit
Columbus
Lexington
Knoxville
Birmingham
Atlanta
Pensacola
Jacksonville
Tampa
Miami
S. Ste Marie
LAKE SUPERIOR
LAKE MICHIGAN
LAKE HURON
LAKE ERIE
LAKE ONTARIO
Buffalo
Pittsburgh
Harrisburg
Philadelphia
New York
Washington
Charleston
Richmond
Norfolk
Raleigh
Hatteras
Columbia
Wilmington
Charleston
Caribou
Burlington
Portland
Concord
Albany
Boston
Hartford
GULF OF MEXICO
OCEAN
800
600
400
200
100
50
0
900
ALASKA
Barrow 1944
Barter Is. 1923
ARCTIC CIRCLE
Ft. Yukon
Fairbanks 1068
Nome 1314
Northway
Anchorage 879
Bethel 1173
Seward
Yakutat 840
Juneau 810
Sitka
Annette 663
Kodiak
St. Paul Is. 1098
Cold Bay 951
Shemya 885
Adak
HAWAII
Lihue
Honolulu
Kahului
Hilo
PUERTO RICO AND VIRGIN ISLANDS
San Juan
Ponce
Alex. Hamilton Fld.
SCALE 1:10 000 000
ALBERS EQUAL AREA PROJECTION - STANDARD PARALLELS 29½° AND 45½°

Figure 4-17



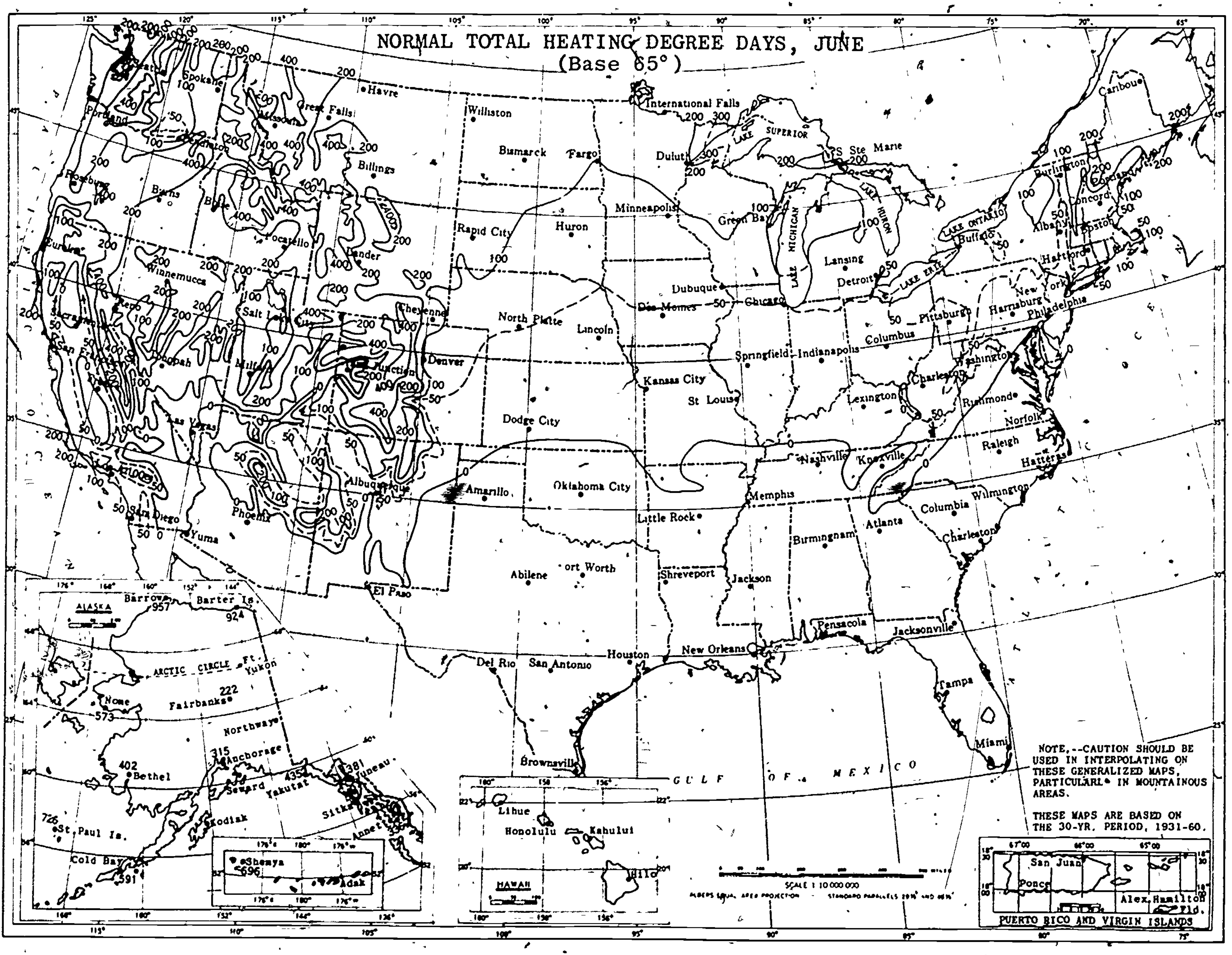
NORMAL TOTAL HEATING DEGREE DAYS, MAY
(Base 65°)

NOTE.--CAUTION SHOULD BE USED IN INTERPOLATING ON THESE GENERALIZED MAPS, PARTICULARLY IN MOUNTAINOUS AREAS.

THESE MAPS ARE BASED ON THE THE 30-YR. PERIOD, 1931-60.

Seattle
Spokane
Portland
Havre
Great Falls
Missoula
Williston
International Falls
Bismarck
Fargo
Duluth
LAKE SUPERIOR
S. Ste Marie
Caribou
Pendleton
Billings
Roseburg
Burns
Boise
Pocatello
Lander
Rapid City
Huron
Minneapolis
Green Bay
LAKE MICHIGAN
LAKE HURON
LAKE ONTARIO
LAKE ERIE
Burlington
Portland
Concord
Albany
Boston
Buffalo
Hartford
Eureka
Winnemucca
Reno
Salt Lake
Cheyenne
North Platte
Lincoln
Des Moines
Dubuque
Chicago
Lansing
Detroit
New York
Harrisburg
Philadelphia
Pittsburgh
Sacramento
San Francisco
Tonopah
Milford
Grand Junction
Denver
Kansas City
Springfield
Indianapolis
Columbus
Washington
Charleston
St. Louis
Lexington
Richmond
Norfolk
Fresno
Las Vegas
Dodge City
Nashville
Knoxville
Raleigh
Hatteras
Los Angeles
San Diego
Yuma
Phoenix
Albuquerque
Amarillo
Oklahoma City
Memphis
Little Rock
Columbia
Wilmington
Birmingham
Atlanta
Charleston
El Paso
Abilene
Fort Worth
Shreveport
Jackson
Pensacola
Jacksonville
Del Rio
San Antonio
Houston
New Orleans
Tampa
Miami
Brownsville
GULF OF MEXICO
ALASKA
Barrow 1445
Barter Is. 1373
ARCTIC CIRCLE
Ft. Yukon
Fairbanks 555
Nome
930
Northway
806 Bethel
592 Anchorage
Seward
Yakutat 632
601 Juneau
Sitka
Annette 505
Kodiak
936 St. Paul Is.
Cold Bay 791
Shemya 837
Adak
HAWAII
Lihue
Honolulu
Kahului
Hilo
PUERTO RICO AND VIRGIN ISLANDS
San Juan
Ponce
Alex. Hamilton Fld.
SCALE 1:10,000,000
ALBERS EQUAL AREA PROJECTION · STANDARD PARALLELS 29½° AND 45½°

Figure 4-18



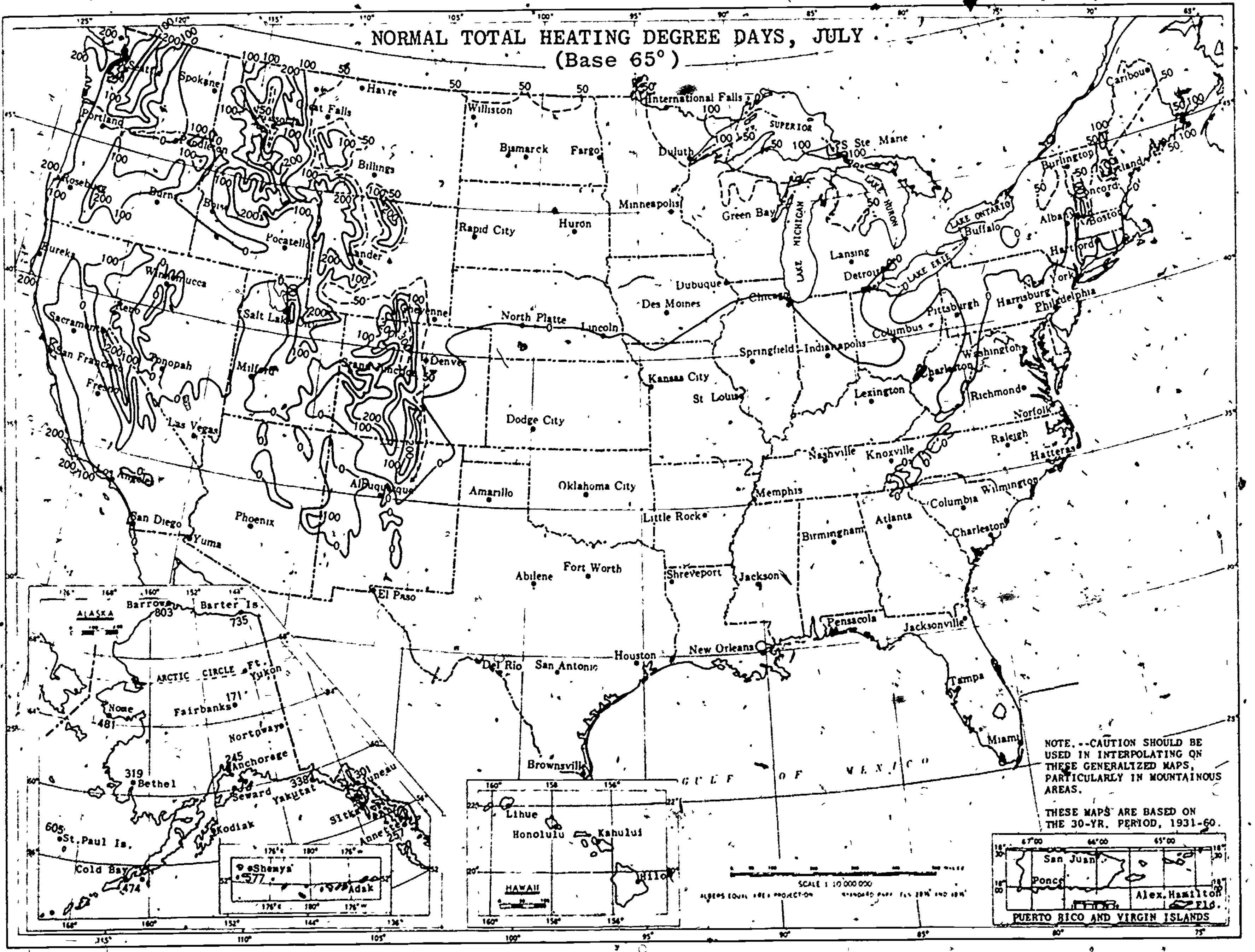
NORMAL TOTAL HEATING DEGREE DAYS, JUNE
(Base 65°)

NOTE.--CAUTION SHOULD BE USED IN INTERPOLATING ON THESE GENERALIZED MAPS, PARTICULARLY IN MOUNTAINOUS AREAS.

THESE MAPS ARE BASED ON THE 30-YR. PERIOD, 1931-60.

Seattle
Spokane
Portland
Pendleton
Roseburg
Burns
Boise
Eureka
Winnemucca
Reno
Sacramento
San Francisco
Tonopah
Milford
Salt Lake City
Pocatello
Lander
Great Falls
Missoula
Havre
Billings
Cheyenne
Denver
Grand Junction
Las Vegas
Los Angeles
San Diego
Yuma
Phoenix
Albuquerque
El Paso
Williston
Bismarck
Fargo
Rapid City
Huron
North Platte
Lincoln
Dodge City
Amarillo
Oklahoma City
Abilene
Fort Worth
Del Rio
San Antonio
Houston
Brownsville
International Falls
Duluth
Minneapolis
Green Bay
Dubuque
Des Moines
Chicago
Kansas City
St Louis
Springfield
Indianapolis
Little Rock
Shreveport
Memphis
Jackson
Nashville
New Orleans
Pensacola
Birmingham
Atlanta
Knoxville
Lexington
Columbus
Lansing
Detroit
S Ste Marie
Lake Superior
Lake Michigan
Lake Huron
Lake Ontario
Lake Erie
Buffalo
Pittsburgh
Charleston
Washington
Richmond
Norfolk
Raleigh
Hatteras
Wilmington
Columbia
Charleston
Jacksonville
Tampa
Miami
Harrisburg
Philadelphia
New York
Hartford
Albany
Burlington
Concord
Portland
Boston
Caribou
Gulf of Mexico
50
100
200
300
400
ALASKA
Barrow 957
Barter Is. 924
Arctic Circle
Ft. Yukon
Fairbanks 222
Nome 573
Northway
Bethel 402
Anchorage 315
Seward
Yakutat 435
Juneau 381
Sitka
Annette
Kodiak
St. Paul Is. 726
Cold Bay 591
Shemya 696
Adak
HAWAII
Lihue
Honolulu
Kahului
Hilo
SCALE 1:10,000,000
ALBERS EQUAL AREA PROJECTION · STANDARD PARALLELS 29½° AND 45½°
PUERTO RICO AND VIRGIN ISLANDS
San Juan
Ponce
Alex. Hamilton Fld.

Figure 4-19

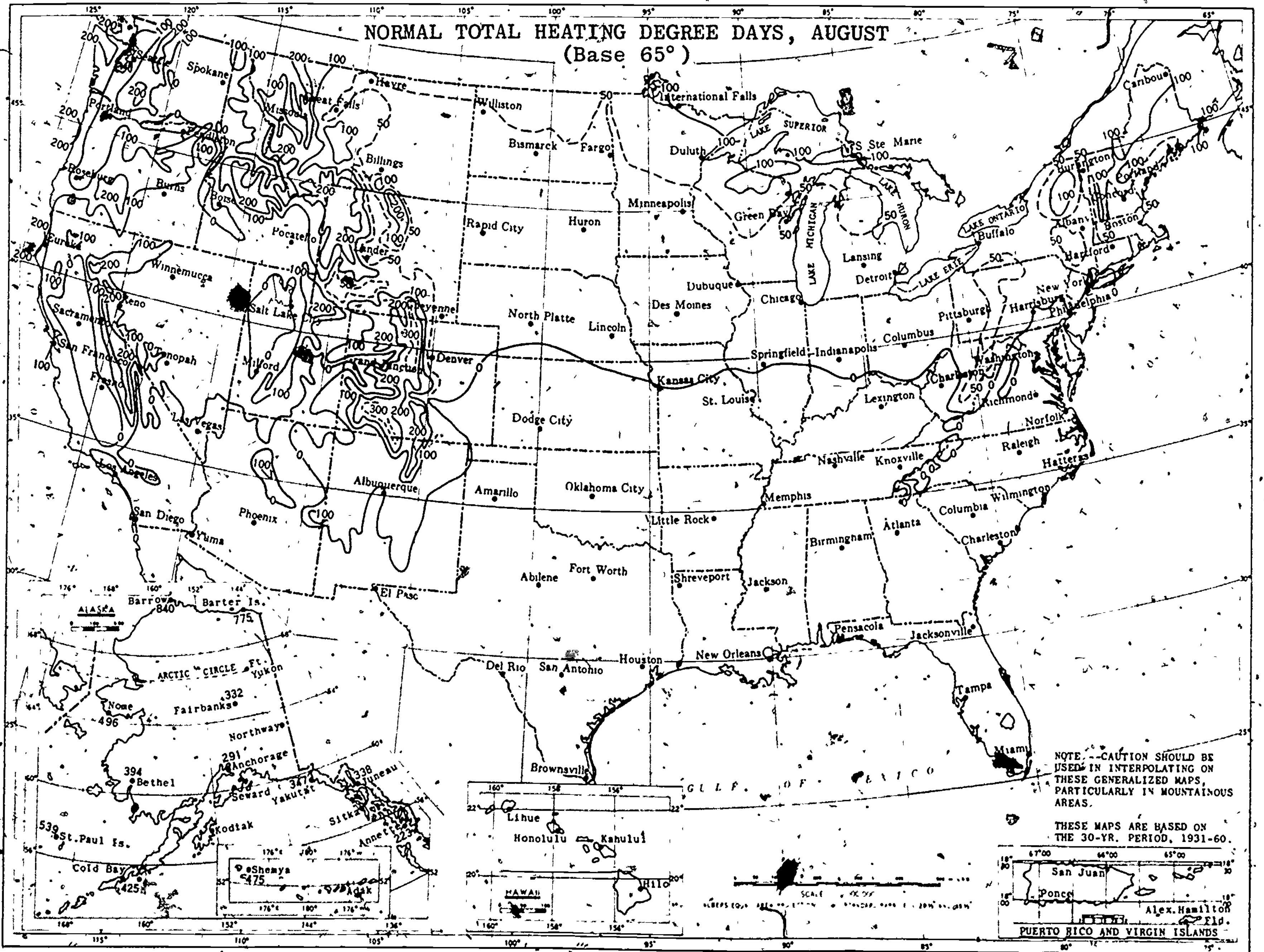

Williston
Bismarck
Fargo
Havre
Great Falls
Billings
Spokane
Seattle
Portland
Pendleton
Roseburg
Burns
Boise
Pocatello
Lander
Eureka
Winnemucca
Reno
Salt Lake City
Sacramento
San Francisco
Tonopah
Fresno
Milford
Las Vegas
Los Angeles
San Diego
Yuma
Phoenix
Grand Junction
Denver
Cheyenne
Albuquerque
El Paso
Rapid City
Huron
North Platte
Lincoln
Dodge City
Amarillo
Oklahoma City
Abilene
Fort Worth
Del Rio
San Antonio
Houston
Brownsville
International Falls
Duluth
Minneapolis
Des Moines
Dubuque
Kansas City
St Louis
Springfield
Chicago
Indianapolis
Green Bay
Lansing
Detroit
Columbus
Lexington
Nashville
Knoxville
Memphis
Little Rock
Shreveport
Jackson
Birmingham
Atlanta
New Orleans
Pensacola
Jacksonville
Tampa
Miami
S Ste Marie
Buffalo
Pittsburgh
Harrisburg
Philadelphia
New York
Hartford
Boston
Albany
Concord
Burlington
Portland
Caribou
Washington
Charleston
Richmond
Norfolk
Raleigh
Hatteras
Wilmington
Columbia
Charleston
LAKE SUPERIOR
LAKE MICHIGAN
LAKE HURON
LAKE ONTARIO
LAKE ERIE
GULF OF MEXICO
ALASKA
Barrow 803
Barter Is. 735
ARCTIC CIRCLE
Ft. Yukon
Fairbanks 171
Nome 481
Northway
Anchorage 245
Bethel 319
Seward
Yakutat 338
Juneau 301
Sitka
Annette 257
Kodiak
St. Paul Is. 605
Cold Bay 474
Shemya 577
Adak
HAWAII
Lihue
Honolulu
Kahului
Hilo
PUERTO RICO AND VIRGIN ISLANDS
San Juan
Ponce
Alex. Hamilton Fld.
SCALE 1:10 000 000
50
100
200

NORMAL TOTAL HEATING DEGREE DAYS, JULY
(Base 65°)

NOTE.--CAUTION SHOULD BE USED IN INTERPOLATING ON THESE GENERALIZED MAPS, PARTICULARLY IN MOUNTAINOUS AREAS.

THESE MAPS ARE BASED ON THE 30-YR. PERIOD, 1931-60.

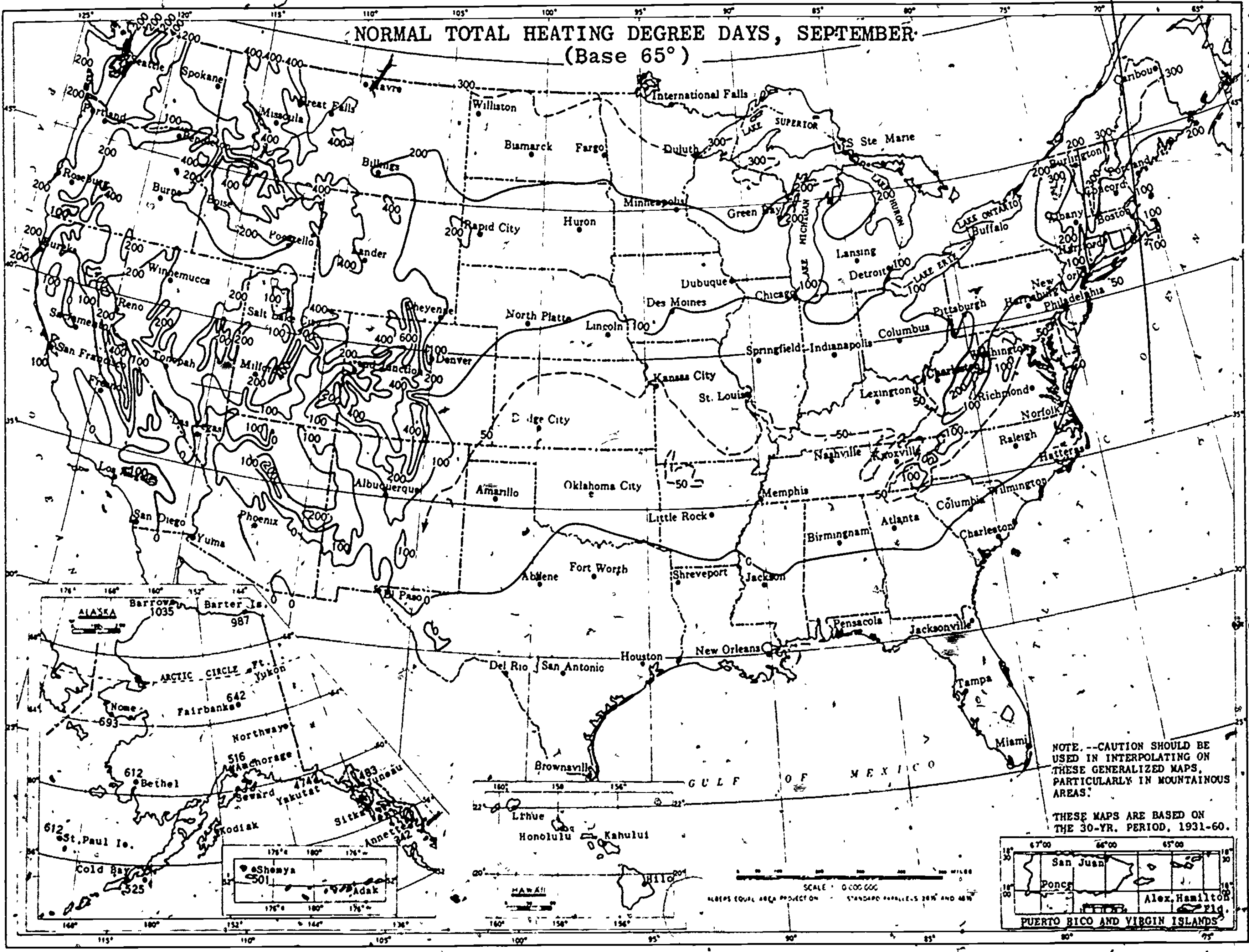
Figure 4-20


NORMAL TOTAL HEATING DEGREE DAYS, AUGUST
(Base 65°)
Seattle
Spokane
Portland
Pendleton
Roseburg
Burns
Boise
Eureka
Winnemucca
Reno
Sacramento
San Francisco
Fresno
Tonopah
Las Vegas
Los Angeles
San Diego
Yuma
Phoenix
Havre
Great Falls
Missoula
Billings
Pocatello
Lander
Salt Lake City
Milford
Cheyenne
Grand Junction
Denver
Albuquerque
El Paso
Williston
Bismarck
Fargo
Rapid City
Huron
North Platte
Lincoln
Dodge City
Amarillo
Oklahoma City
Abilene
Fort Worth
Del Rio
San Antonio
Houston
Brownsville
International Falls
Duluth
Minneapolis
Des Moines
Kansas City
St. Louis
Little Rock
Shreveport
Memphis
Jackson
New Orleans
Green Bay
Dubuque
Chicago
Springfield
Indianapolis
Lansing
Detroit
Columbus
Lexington
Nashville
Knoxville
Birmingham
Atlanta
Pensacola
Jacksonville
Tampa
Miami
S Ste Marie
Buffalo
Pittsburgh
Harrisburg
Charleston
Washington
Richmond
Norfolk
Raleigh
Hatteras
Wilmington
Columbia
Charleston
New York
Philadelphia
Hartford
Albany
Boston
Burlington
Concord
Portland
Caribou
LAKE SUPERIOR
LAKE MICHIGAN
LAKE HURON
LAKE ONTARIO
LAKE ERIE
GULF OF MEXICO
ALASKA
Barrow 840
Barter Is. 775
ARCTIC CIRCLE
Ft. Yukon
Fairbanks 332
Nome 496
Northway
Anchorage 291
Bethel 394
Seward
Yakutat 347
Juneau 338
Sitka
Annette
Kodiak
St. Paul Is. 539
Cold Bay 425
Shemya 475
Adak
HAWAII
Lihue
Honolulu
Kahului
Hilo
PUERTO RICO AND VIRGIN ISLANDS
San Juan
Ponce
Alex. Hamilton Fld.
0
50
100
200
300

NOTE.--CAUTION SHOULD BE USED IN INTERPOLATING ON THESE GENERALIZED MAPS, PARTICULARLY IN MOUNTAINOUS AREAS.

THESE MAPS ARE BASED ON THE 30-YR. PERIOD, 1931-60.

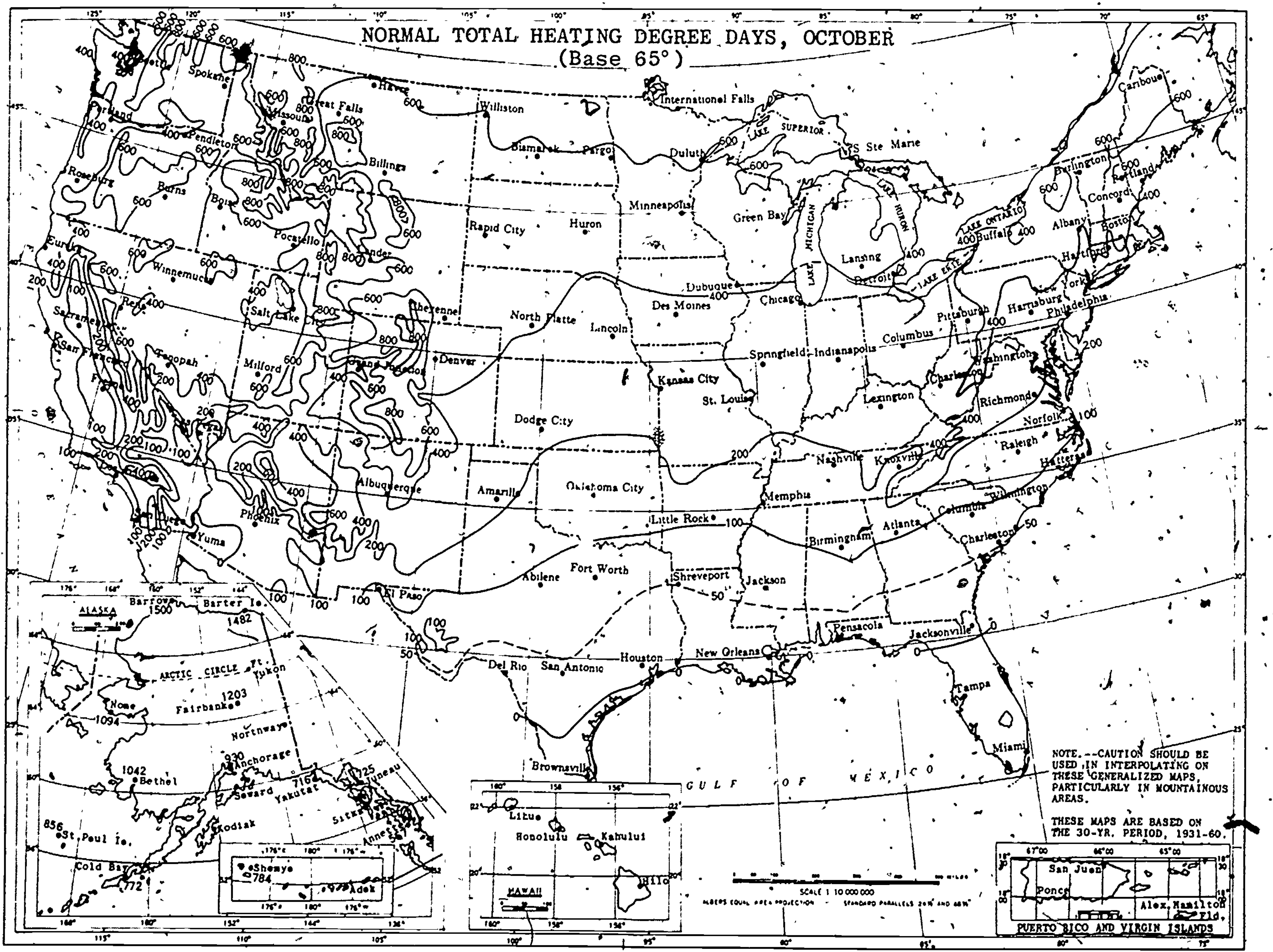
Figure 4-21


NORMAL TOTAL HEATING DEGREE DAYS, SEPTEMBER
(Base 65°)
Seattle
Spokane
Portland
Havre
Great Falls
Missoula
Williston
Bismarck
Fargo
International Falls
Duluth
LAKE SUPERIOR
S Ste Marie
Caribou
Burlington
Portland
Concord
Albany
Boston
Hartford
Buffalo
LAKE ONTARIO
LAKE HURON
LAKE MICHIGAN
LAKE ERIE
Green Bay
Minneapolis
Huron
Rapid City
Billings
Boise
Burns
Roseburg
Pocatello
Lander
Eureka
Winnemucca
Reno
Sacramento
San Francisco
Fresno
Tonopah
Salt Lake City
Milford
Cheyenne
Grand Junction
Denver
North Platte
Lincoln
Des Moines
Dubuque
Chicago
Lansing
Detroit
Pittsburgh
Harrisburg
New York
Philadelphia
Columbus
Springfield
Indianapolis
Kansas City
St. Louis
Lexington
Charleston
Washington
Richmond
Norfolk
Raleigh
Hatteras
Nashville
Knoxville
Dodge City
Las Vegas
Los Angeles
San Diego
Yuma
Phoenix
Albuquerque
Amarillo
Oklahoma City
Little Rock
Memphis
Birmingham
Atlanta
Columbia
Wilmington
Charleston
Abilene
Fort Worth
Shreveport
Jackson
El Paso
Del Rio
San Antonio
Houston
New Orleans
Pensacola
Jacksonville
Tampa
Miami
Brownsville
GULF OF MEXICO
ALASKA
Barrow 1035
Barter Is. 987
ARCTIC CIRCLE
Ft. Yukon
Fairbanks 642
Nome 693
Northway
Anchorage 516
Bethel 612
Seward
Yakutat 474
Juneau 483
Sitka
Annette 342
Kodiak
St. Paul Is. 612
Cold Bay 525
Shemya 501
Adak
HAWAII
Lihue
Honolulu
Kahului
Hilo
PUERTO RICO AND VIRGIN ISLANDS
San Juan
Ponce
Alex. Hamilton Fld.
SCALE 1:10,000,000
ALBERS EQUAL AREA PROJECTION — STANDARD PARALLELS 29½° AND 45½°
50
100
200
300
400
600

NOTE.--CAUTION SHOULD BE USED IN INTERPOLATING ON THESE GENERALIZED MAPS, PARTICULARLY IN MOUNTAINOUS AREAS.

THESE MAPS ARE BASED ON THE 30-YR. PERIOD, 1931-60.

Figure 4-22



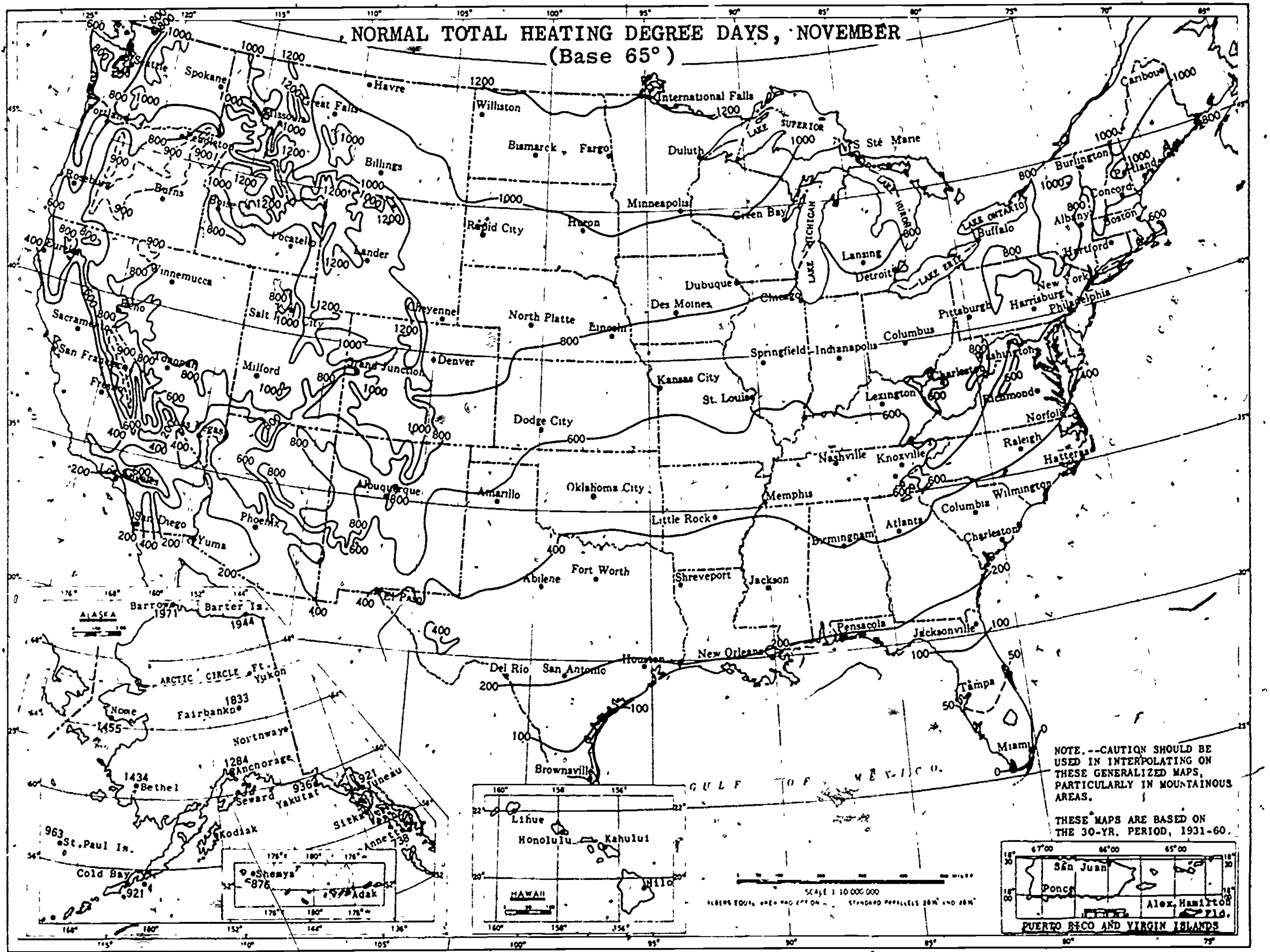
NORMAL TOTAL HEATING DEGREE DAYS, OCTOBER
(Base 65°)

Seattle
Spokane
Portland
Pendleton
Roseburg
Burns
Boise
Eureka
Winnemucca
Reno
Sacramento
San Francisco
Tonopah
Fresno
Los Angeles
San Diego
Yuma
Phoenix
Milford
Salt Lake City
Pocatello
Lander
Great Falls
Missoula
Havre
Billings
Cheyenne
Denver
Grand Junction
Albuquerque
El Paso
Williston
Bismarck
Fargo
Rapid City
Huron
North Platte
Lincoln
Dodge City
Amarillo
Oklahoma City
Abilene
Fort Worth
Del Rio
San Antonio
Houston
Brownsville
International Falls
Duluth
Minneapolis
Des Moines
Dubuque
Kansas City
St. Louis
Little Rock
Shreveport
Jackson
New Orleans
Memphis
Nashville
Knoxville
Birmingham
Atlanta
Pensacola
Jacksonville
Tampa
Miami
Green Bay
Chicago
Springfield
Indianapolis
Lansing
Detroit
Columbus
Lexington
Charleston
Pittsburgh
Harrisburg
Philadelphia
New York
Washington
Richmond
Norfolk
Raleigh
Hatteras
Wilmington
Columbia
Charleston
Buffalo
Albany
Hartford
Boston
Concord
Burlington
Portland
Caribou
S Ste Marie
LAKE SUPERIOR
LAKE MICHIGAN
LAKE HURON
LAKE ERIE
LAKE ONTARIO
GULF OF MEXICO
ALASKA
Barrow 1500
Barter Is. 1482
ARCTIC CIRCLE
Ft. Yukon
Fairbanks 1203
Nome
1094
Northway
Anchorage 930
Bethel 1042
Seward
Yakutat 716
Juneau 725
Sitka
Annette
Kodiak
St. Paul Is. 856
Cold Bay 772
Shemya 784
Adak
HAWAII
Lihue
Honolulu
Kahului
Hilo
San Juan
Ponce
Alex. Hamilton Fld.
PUERTO RICO AND VIRGIN ISLANDS
600
800
400
200
100
50
0
SCALE 1:10 000 000
ALBERS EQUAL AREA PROJECTION - STANDARD PARALLELS 29½° AND 45½°

NOTE.--CAUTION SHOULD BE USED IN INTERPOLATING ON THESE GENERALIZED MAPS, PARTICULARLY IN MOUNTAINOUS AREAS.

THESE MAPS ARE BASED ON THE 30-YR. PERIOD, 1931-60.

Figure 4-23



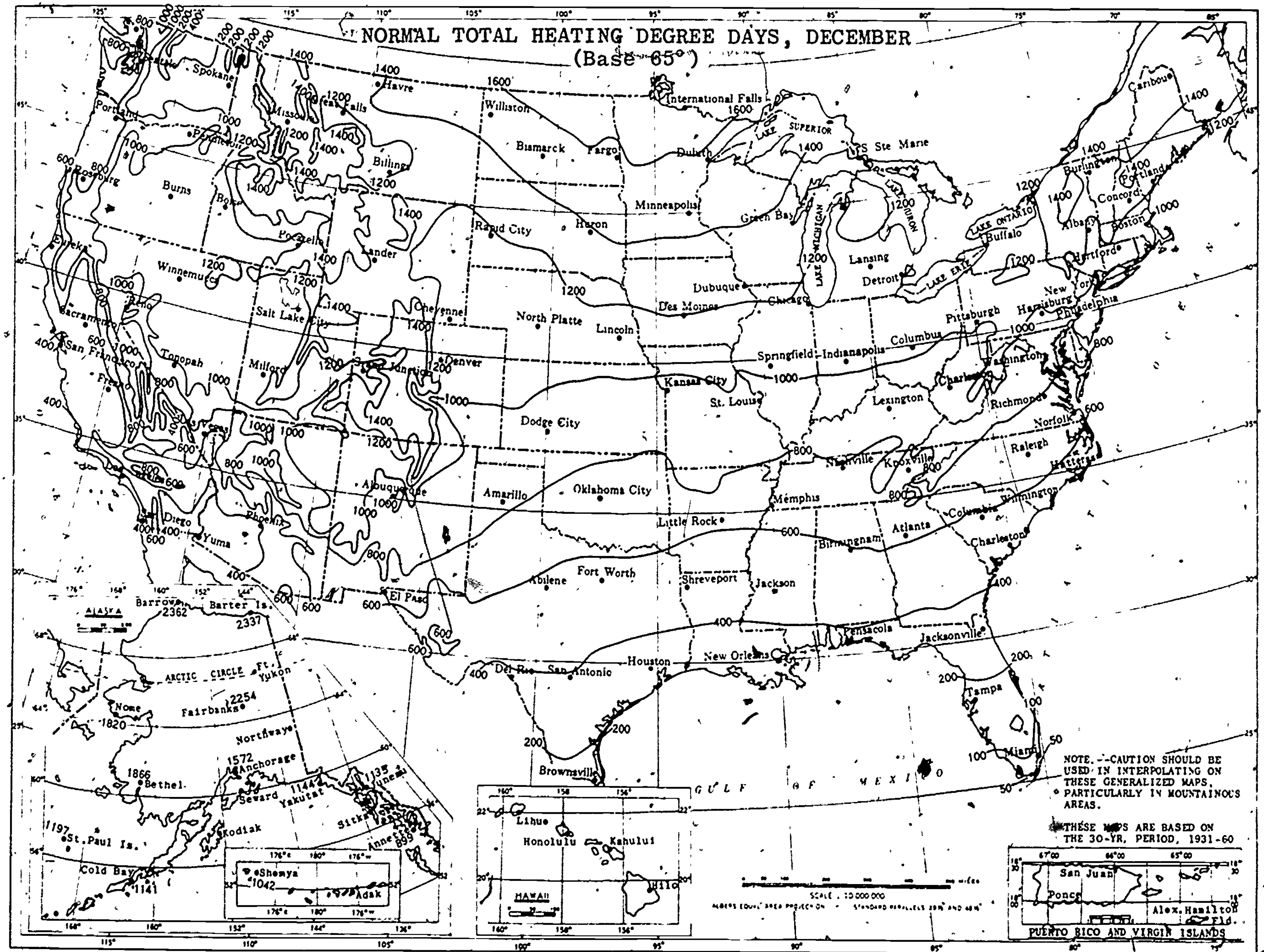
NORMAL TOTAL HEATING DEGREE DAYS, NOVEMBER
(Base 65°)

Seattle
Spokane
Portland
Roseburg
Burns
Pendleton
Eureka
Winnemucca
Reno
Sacramento
San Francisco
Fresno
Tonopah
Las Vegas
Los Angeles
San Diego
Yuma
Phoenix
Boise
Pocatello
Missoula
Great Falls
Havre
Billings
Lander
Salt Lake City
Milford
Grand Junction
Cheyenne
Denver
Albuquerque
El Paso
Williston
Bismarck
Fargo
Rapid City
Huron
North Platte
Lincoln
Dodge City
Amarillo
Oklahoma City
Abilene
Fort Worth
Del Rio
San Antonio
Houston
Brownsville
International Falls
Duluth
Minneapolis
Green Bay
Dubuque
Des Moines
Kansas City
St. Louis
Springfield
Indianapolis
Chicago
Lansing
Detroit
Columbus
Lexington
Nashville
Knoxville
Memphis
Little Rock
Shreveport
Jackson
Birmingham
Atlanta
New Orleans
Pensacola
Jacksonville
Tampa
Miami
S Ste Marie
LAKE SUPERIOR
LAKE MICHIGAN
LAKE HURON
LAKE ONTARIO
LAKE ERIE
Buffalo
Pittsburgh
Harrisburg
Philadelphia
New York
Hartford
Boston
Albany
Concord
Portland
Burlington
Caribou
Washington
Charleston
Richmond
Norfolk
Raleigh
Hatteras
Wilmington
Columbia
Charleston
GULF OF MEXICO
1200
1000
900
800
600
400
200
100
50
0
ALASKA
Barrow 1971
Barter Is. 1944
ARCTIC CIRCLE
Ft. Yukon
Nome
1455
Fairbanks 1833
Northway
Anchorage 1284
Bethel 1434
Seward
Yakutat 936
Juneau 921
Sitka
Annette 738
Kodiak
St. Paul Is. 963
Cold Bay 921
Shemya 876
Adak
HAWAII
Lihue
Honolulu
Kahului
Hilo
SCALE 1:10 000 000
ALBERS EQUAL AREA PROJECTION STANDARD PARALLELS 29½° AND 45½°
San Juan
Ponce
Alex. Hamilton Fld.
PUERTO RICO AND VIRGIN ISLANDS

NOTE.--CAUTION SHOULD BE USED IN INTERPOLATING ON THESE GENERALIZED MAPS, PARTICULARLY IN MOUNTAINOUS AREAS.

THESE MAPS ARE BASED ON THE 30-YR. PERIOD, 1931-60.

Figure 4-24



NORMAL TOTAL HEATING DEGREE DAYS, DECEMBER (Base 65°)

NOTE.--CAUTION SHOULD BE USED IN INTERPOLATING ON THESE GENERALIZED MAPS, PARTICULARLY IN MOUNTAINOUS AREAS.

THESE MAPS ARE BASED ON THE 30-YR. PERIOD, 1931-60

Seattle
Spokane
Portland
Pendleton
Roseburg
Burns
Boise
Eureka
Winnemucca
Reno
Sacramento
San Francisco
Fresno
Tonopah
Milford
Salt Lake City
Pocatello
Missoula
Great Falls
Havre
Helena
Billings
Lander
Cheyenne
Denver
Grand Junction
Las Vegas
Los Angeles
San Diego
Yuma
Phoenix
Albuquerque
El Paso
Williston
Bismarck
Fargo
Rapid City
Huron
North Platte
Lincoln
Dodge City
Amarillo
Oklahoma City
Abilene
Fort Worth
Del Rio
San Antonio
Houston
Brownsville
International Falls
Duluth
Minneapolis
Des Moines
Dubuque
Kansas City
St. Louis
Springfield
Little Rock
Shreveport
Jackson
New Orleans
Memphis
Nashville
Birmingham
Atlanta
Pensacola
Jacksonville
Tampa
Miami
Chicago
Green Bay
Sault Ste Marie
Lansing
Detroit
Indianapolis
Columbus
Lexington
Knoxville
Pittsburgh
Charleston
Washington
Richmond
Norfolk
Raleigh
Hatteras
Wilmington
Columbia
Charleston
Buffalo
Albany
Burlington
Concord
Portland
Caribou
Boston
Hartford
New York
Harrisburg
Philadelphia
LAKE SUPERIOR
LAKE MICHIGAN
LAKE HURON
LAKE ONTARIO
LAKE ERIE
GULF OF MEXICO
1600
1400
1200
1000
800
600
400
200
100
50
ALASKA
Barrow 2362
Barter Is. 2337
ARCTIC CIRCLE
Ft. Yukon
Fairbanks 2254
Nome 1820
Northway
Anchorage 1572
Bethel 1866
Seward
Yakutat 1144
Juneau 1135
Sitka
Annette 899
Kodiak
St. Paul Is. 1197
Cold Bay 1141
Shemya 1042
Adak
HAWAII
Lihue
Honolulu
Kahului
Hilo
PUERTO RICO AND VIRGIN ISLANDS
San Juan
Ponce
Alex. Hamilton Fld.
SCALE 1:10,000,000
ALBERS EQUAL AREA PROJECTION STANDARD PARALLELS 29½° AND 45½°

Table 4-5. Heating Load Data*

	NORMAL TOTAL HEATING DEGREE DAYS (Base 65^0)													Design T_0 ^{0}F	
STATE AND STATION	JULY	AUG	SEPT	OCT	NOV	DEC	JAN	FEB	MAR	APR	MAY	JUNE	ANNUAL	WIN.†	SUMM.Π
ALA: Birmingham	0	0	6	93	363	555	592	462	363	108	9	0	2551	19	97
Huntsville	0	0	12	127	426	663	694	557	434	138	19	0	3070	13	97
Mobile	0	0	0	22	213	357	415	300	211	42	0	0	1560	26	95
Montgomery	0	0	0	68	330	527	543	417	[illegible]16	90	0	0	2291	22	98
ALASKA: Anchorage	245	291	516	930	1284	1572	1631	1316	1293	879	592	315	10864	-25	73
Annette	242	208	327	567	738	899	949	837	843	648	490	321	7069		
Barrow	803	840	1035	1500	1971	2362	2517	2332	2468	1944	1445	957	20174	-45	-58
Barter Is.	735	775	987	1482	1944	2337	2536	2369	2477	1923	1373	924	19862		
Bethel	319	394	612	1042	1434	1866	1903	1590	1655	1173	806	402	13196		
Cold Bay	474	425	525	772	918	1122	1153	1036	1122	951	791	591	9880		
Cordova	366	391	522	781	1017	1221	1299	1086	1113	864	660	444	9764		
Fairbanks	171	332	642	1203	1833	2254	2359	1901	1739	1068	555	222	14279	-53	82
Juneau	301	338	483	725	921	1135	1237	1070	1073	810	601	381	9075	- 7	75
King Salmon	313	322	513	908	1290	1606	1600	1333	1411	966	673	408	11343		
Kotzebue	381	446	723	1249	1728	2127	2192	1932	2080	1554	1057	636	16105		
McGrath	208	338	633	1184	1791	2232	2294	1817	1758	1122	648	258	14283		
Nome	481	496	693	1094	1455	1820	1879	1666	1770	1314	930	573	14171	-32	66
Saint Paul	605	539	612	862	963	1197	1228	1168	1265	1098	936	726	11199		
Shemya	577	475	501	784	876	1042	1045	958	1011	885	837	696	9687		
Yakutat	338	347	474	716	936	1144	1169	1019	1042	840	632	435	9092		
ARIZ: Flagstaff	46	68	201	558	867	1073	1169	991	911	651	437	180	7152	0	84
Phoenix	0	0	0	22	234	415	474	328	217	75	0	0	1765	31	108
Prescott	0	0	27	245	579	797	865	711	605	360	158	15	4362	15	96
Tucson	0	0	0	25	231	406	471	344	242	75	6	0	1800	29	105
Winslow	0	0	6	245	711	1008	1054	770	601	291	96	0	4782	9	97
Yuma	0	0	0	0	148	319	363	228	130	29	0	0	1217	37	111
ARK: Fort Smith	0	0	12	127	450	704	781	596	456	144	22	0	3292	15	101
Little Rock	0	0	9	127	465	716	756	577	434	126	9	0	3219	19	99
Texarkana	0	0	0	78	345	561	626	468	350	105	0	0	2533	22	99
CALIF: Bakersfield	0	0	0	37	282	502	546	364	267	105	19	0	2122	31	103
Bishop	0	0	42	248	576	797	874	666	539	306	143	36	4227		
Blue Canyon	34	50	120	347	579	766	865	781	791	582	397	195	5507		
Burbank	0	0	6	43	177	301	366	277	239	138	81	18	1646	36	97

*From Climatic Atlas of the United States, U.S. Department of Commerce, Env. Sci. Serv. Adm. June 1968

†From Table 1, Chapter 33, ASHRAE Handbook of Fundamentals 1972 (99% of time warmer than this temperature)

ΠFrom Table 1, Chapter 33, ASHRAE Handbook of Fundamentals 1972 (1% of time dry-bulb temperature is greater)

NORMAL TOTAL HEATING DEGREE DAYS (Base 65^{0})														Design T_{o} ^{o}F	
STATE AND STATION	JULY	AUG	SEPT	OCT	NOV	DEC	JAN	FEB	MAR	APR	MAY	JUNE	ANNUAL	WIN.	SUMM.
CALIF: Eureka	270	257	258	329	414	499	546	470	505	438	372	285	4643	32	67
Fresno	0	0	0	78	339	558	586	406	319	150	56	0	2492	28	101
Long Beach	0	0	12	40	156	288	375	297	267	168	90	18	1711	36	87
Los Angeles	28	22	42	78	180	291	372	302	288	219	158	81	2061	42	94
Mt. Shasta	25	34	123	406	696	902	983	784	738	525	347	159	5722		
Oakland	53	50	45	127	309	481	527	400	353	255	180	90	2870	35	85
Point Arguello	202	186	162	205	291	400	474	392	403	339	298	243	3595		
Red Bluff	0	0	0	53	318	555	605	428	341	168	47	0	2515		
Sacramento	0	0	12	81	363	577	614	442	360	216	102	6	2773	30	100
Sandberg	0	0	30	202	480	691	778	661	620	426	264	57	4209		
San Diego	6	0	15	37	123	251	313	249	202	123	84	36	1439	42	86
San Francisco	81	78	60	143	306	462	508	395	363	279	214	126	3015	42	80
Santa Catalina	16	0	9	50	165	279	353	308	326	249	192	105	2052		
Santa Maria	99	93	96	146	270	391	459	370	363	282	233	165	2967	32	85
COLO: Alamosa	65	99	279	639	1065	1420	1476	1162	1020	696	440	168	8529	-17	84
Colorado Springs	9	25	132	456	825	1032	1128	938	893	582	319	84	6423	- 1	90
Denver	6	9	117	428	819	1035	1132	938	887	558	288	66	6283	- 2	92
Grand Junction	0	0	30	313	786	1113	1209	907	729	387	146	21	5641	8	96
Pueblo	0	0	54	326	750	986	1085	871	772	429	174	15	5462	- 5	96
CONN: Bridgeport	0	0	66	307	615	986	1079	966	853	510	208	27	5617	4	90
Hartford	0	6	99	372	711	1119	1209	1061	899	495	177	24	6172	1	90
New Haven	0	12	87	347	648	1011	1097	991	871	543	245	45	5897	5	88
DEL: Wilmington	0	0	51	270	588	927	980	874	735	387	112	6	4930	12	93
FLA: Apalachicola	0	0	0	16	153	319	347	260	180	33	0	0	1308		
Daytona Beach	0	0	0	0	75	211	248	190	140	15	0	0	879	32	94
Fort Myers	0	0	0	0	24	109	146	101	62	0	0	0	442	38	94
Jacksonville	0	0	0	12	144	310	332	246	174	21	0	0	1239	29	96
Key West	0	0	0	0	0	28	40	31	9	0	0	0	108	55	90
Lakeland	0	0	0	0	57	164	195	146	99	0	0	0	661	35	95
Miami Beach	0	0	0	0	0	40	56	36	9	0	0	0	141	45	91
Orlando	0	0	0	0	72	198	220	165	105	6	0	0	766	33	96
Pensacola	0	0	0	19	195	353	400	277	183	36	0	0	1463	29	92
Tallahassee	0	0	0	28	198	360	375	286	202	36	0	0	1485	25	96

	NORMAL TOTAL HEATING DEGREE DAYS (Base 65^0)													Design T_0 ^{0}F	
STATE AND STATION	JULY	AUG	SEPT	OCT	NOV	DEC	JAN	FEB	MAR	APR	MAY	JUNE	ANNUAL	WIN.	SUMM.
FLA: Tampa	0	0	0	0	60	171	202	148	102	0	0	0	683	36	92
West Palm Beach	0	0	0	0	6	65	87	64	31	0	0	0	253	40	92
GA: Athens	0	0	12	115	405	632	642	529	431	141	22	0	2929	17	96
Atlanta	0	0	18	127	414	626	639	529	437	168	25	0	2983	18	95
Augusta	0	0	0	78	333	552	549	445	350	90	0	0	2397	20	98
Columbus	0	0	0	87	333	543	552	434	338	96	0	0	2383	23	98
Macon	0	0	0	71	297	502	505	403	295	63	0	0	2136	23	98
Rome	0	0	24	161	474	701	710	577	468	177	34	0	3326	16	97
Savannah	0	0	0	47	246	437	437	353	254	45	0	0	1819	24	96
Thomasville	0	0	0	25	198	366	394	305	208	33	0	0	1529		
IDAHO: Boise	0	0	132	415	792	1017	1113	854	722	438	245	81	5809	4	96
Idaho Falls 46W	16	34	270	623	1056	1370	1538	1249	1085	651	391	192	8475		
Idaho Falls 42NW	16	40	282	648	1107	1432	1600	1291	1107	657	388	192	8760		
Lewiston	0	0	123	403	756	933	1063	815	694	426	239	90	5542	6	98
Pocatello	0	0	172	493	900	1166	1324	1058	905	555	319	141	7033	- 8	94
ILL: Cairo	0	0	36	164	513	791	856	680	539	195	47	0	3821		
Chicago	0	0	81	326	753	1113	1209	1044	890	480	211	48	6155	- 3	94
Moline	0	9	99	335	774	1181	1314	1100	918	450	189	39	6408	- 7	94
Peoria	0	6	87	326	759	1113	1218	1025	849	426	183	33	6025	- 2	94
Rockford	6	9	114	400	837	1221	1333	1137	961	516	236	60	6830	- 7	92
Springfield	0	0	72	291	696	1023	1135	935	769	354	136	18	5429	- 1	95
IND: Evansville	0	0	66	220	606	896	955	767	620	237	[illegible]	0	4435	6	96
Fort Wayne	0	9	105	378	783	1135	1178	1028	890	471	[illegible]	39	6205	0	93
Indianapolis	0	0	90	316	723	1051	1113	949	809	432	177	39	5699	0	93
South Bend	[illegible]	6	111	372	777	1125	1221	1070	933	525	239	60	6439	- 2	92
IOWA: Burlington	0	0	93	322	768	1135	1259	1042	859	426	177	33	6114	- 4	95
Des Moines	0	9	99	363	837	1231	1398	1163	967	489	211	39	6808	- 7	95
Dubuque	12	31	156	450	906	1287	1420	1204	1026	546	260	78	7376	-11	92
Sioux City	0	9	108	369	867	1240	1435	1198	989	483	214	39	6951	-10	96
Waterloo	12	19	138	428	909	1296	1460	1221	1023	531	229	54	7320	-12	91

NORMAL TOTAL HEATING DEGREE DAYS (Base 65^o)														Design T_o ^{o}F	
STATE AND STATION	JULY	AUG	SEPT	OCT	NOV	DEC	JAN	FEB	MAR	APR	MAY	JUNE	ANNUAL	WIN.	SUMM.
KANSAS: Concordia	0	0	57	276	706	1023	[illegible]63	935	781	372	149	18	5479		
Dodge City	0	0	33	251	666	939	1051	840	719	354	124	9	4986	3	99
Goodland	0	6	81	381	810	1073	1166	955	884	507	236	42	6141	- 2	99
Topeka	0	0	57	270	672	980	1122	893	722	330	124	12	5182	3	99
Wichita	0	0	33	229	618	905	1023	804	645	270	87	6	4620	5	102
KY: Covington	0	0	75	291	669	983	1035	893	756	390	149	24	5265	3	93
Lexington	0	0	54	239	609	902	946	818	685	325	105	0	4683	6	94
Louisville	0	0	54	248	609	890	930	818	682	315	105	9	4660	8	96
LA: Alexandria	0	0	0	56	273	431	471	361	260	69	0	0	1921	25	97
Baton Rouge	0	0	0	31	216	369	409	294	208	33	0	0	1560	25	96
Burrwood	0	0	0	0	96	214	298	218	171	27	0	0	1024		
Lake Charles	0	0	0	19	210	341	381	274	195	39	0	0	1459	29	95
New Orleans	0	0	0	19	192	322	363	258	192	39	0	0	1385	32	93
Shreveport	0	0	0	47	297	477	552	426	304	81	0	0	2184	22	99
MAINE: Caribou	78	115	336	682	1044	1535	1690	1470	1308	858	468	183	9767	-18	85
Portland	12	53	195	508	807	1215	1339	1182	1042	675	372	111	7511	- 5	88
MD: Baltimore	0	0	48	264	585	905	936	820	679	327	90	0	4654	16	94
Frederick	0	0	66	307	624	955	995	876	741	384	127	12	5087	7	94
MASS: Blue Hill Obsy	0	22	108	381	690	1085	1178	1053	936	579	267	69	6368		
Boston	0	9	60	316	603	983	1088	972	846	513	208	36	5634	6	91
Nantucket	12	22	93	332	573	896	992	941	896	621	384	129	5891		
Pittsfield	25	59	219	524	831	1231	1339	1196	1063	660	326	105	7578	- 1	86
Worcester	6	34	147	450	774	1172	1271	1123	998	612	304	78	6969	1	89
MICH: Alpena	68	105	273	580	912	1268	1404	1299	1218	777	446	156	8506	- 5	87
Detroit (City)	0	0	87	360	738	1088	1181	1058	936	522	220	42	6232	4	92
Escanaba	59	87	243	539	924	1293	1445	1296	1203	777	456	159	8481	- 7	82
Flint	16	40	159	465	843	1212	1330	1198	1066	639	319	90	7377	- 1	89
Grand Rapids	9	28	135	434	804	1147	1259	1134	1011	549	279	75	6894	2	91
Lansing	6	22	138	431	813	1163	1262	1142	1011	579	273	69	6909	2	89
Marquette	59	81	240	527	936	1268	1411	1268	1187	771	468	177	8393	- 8	88
Muskegon	12	28	120	400	762	1088	1209	1100	995	594	310	78	6696	4	87
Sault Ste. Marie	96	105	279	580	951	1367	1525	1380	1277	810	477	201	9048	-12	83

NORMAL TOTAL HEATING DEGREE DAYS (Base 65^{o})														Design T_O ^{o}F	
STATE AND STATION	JULY	AUG	SEPT	OCT	NOV	DEC	JAN	FEB	MAR	APR	MAY	JUNE	ANNUAL	WIN.	SUMM.
MINN: Duluth	71	109	330	632	1131	1581	1745	1518	1355	840	490	198	10000	-19	85
Internat'l Falls	71	112	363	701	1236	1724	1919	1621	1414	828	443	174	10606	-29	86
Minneapolis	22	31	189	505	1014	1454	1631	1380	1166	621	288	81	8382	-14	92
Rochester	25	34	186	474	1005	1438	1593	1366	1150	630	301	93	8295	-17	90
Saint Cloud	28	47	225	549	1065	1500	1702	1445	1221	666	326	105	8879	-20	90
MISS: Jackson	0	0	0	65	315	502	546	414	310	87	0	0	2239	21	98
Meridian	0	0	0	81	339	518	5[illegible]3	417	310	81	0	0	2289	20	97
Vicksburg	0	0	0	53	279	462	512	384	282	69	0	0	2041	23	97
MO: Columbia	0	0	54	251	651	967	1076	874	716	324	121	12	5046	2	97
Kansas	0	0	39	220	612	905	1032	818	682	294	109	0	4711	4	100
St. Joseph	0	6	60	285	708	1039	1172	949	769	348	133	15	5484	- 1	97
St. Louis	0	0	60	251	627	936	1026	848	704	312	121	15	4900	7	96
Springfield	0	0	45	223	600	877	973	781	660	291	105	6	4561	5	97
MONT: Billings	6	15	186	487	897	1135	1296	1100	970	570	285	102	7049	-10	94
Glasgow	31	47	270	608	1104	1466	1711	1439	1187	648	335	150	8996	-25	96
Great Falls	28	53	258	543	921	1169	1349	1154	1063	642	384	186	7750	-20	91
Havre	28	53	306	595	1065	1367	1584	1364	1181	657	338	162	8700	-22	91
Helena	31	59	294	601	1002	1265	1438	1170	1042	651	381	195	8129	-17	90
Kalispell	50	99	321	654	1020	1240	1401	1134	1029	639	397	207	8191	- 7	88
Miles City	6	6	174	502	972	1296	1504	1252	1057	579	276	99	7723	-19	97
Missoula	34	74	303	651	1035	1287	1420	1120	970	621	391	219	8125	- 7	92
NEBR: Grand Island	0	6	108	381	834	1172	1314	1089	908	462	211	45	6530	- 6	98
Lincoln	0	6	75	301	726	1066	1237	1016	834	402	171	30	5864	- 4	100
Norfolk	9	0	111	397	873	1234	1414	1179	983	498	233	48	6979	-11	97
North Platte	0	6	123	440	885	1166	1271	1039	930	519	248	57	6684	- 6	97
Omaha	0	12	105	357	828	1175	1355	1126	939	465	208	42	6612	- 5	97
Scottsbluff	0	0	138	459	876	1128	1231	1008	921	552	285	75	6673	- 8	96
Valentine	9	12	165	493	942	1237	1395	1176	1045	579	288	84	7425		
NEV: Elko	9	34	225	561	924	1197	1314	1036	911	621	409	192	7433	-13	94
Ely	28	43	234	592	939	1184	1308	1075	977	672	456	225	7733	- 6	90
Las Vegas	0	0	0	78	387	617	688	487	335	111	6	0	2709	23	108
Reno	43	87	204	490	801	1026	1073	823	729	510	357	189	6332	12	94
Winnemucca	0	34	210	536	876	1091	1172	916	837	573	363	153	6761	1	97

NORMAL TOTAL HEATING DEGREE DAYS (Base 65°)														Design T_o °F	
STATE AND STATION	JULY	AUG	SEPT	OCT	NOV	DEC	JAN	FEB	MAR	APR	MAY	JUNE	ANNUAL	WIN.	SUMM.
NH: Concord	6	50	177	505	822	1240	1358	1184	1032	636	298	75	7383	-11	91
Mt. Wash, Obsy.	493	536	720	1057	1341	1742	1820	1663	1652	1260	930	603	13817		
NJ: Atlantic City	0	0	39	251	549	880	936	848	741	420	133	15	4812	14	91
Newark	0	0	30	248	573	921	983	876	729	381	118	0	4859	11	94
Trenton	0	0	57	264	576	924	989	885	753	399	121	12	4980	12	92
NM: Albuquerque	0	0	12	229	642	868	930	703	595	288	81	0	4348	14	96
Clayton	0	6	66	310	699	899	986	812	747	429	183	21	5158		
Raton	9	28	126	431	825	1048	1116	904	834	543	301	63	6228	- 2	92
Roswell	0	0	18	202	573	806	840	641	481	201	31	0	3793	16	101
Silver City	0	0	6	183	525	729	791	605	581	261	87	0	3705	14	95
NY: Albany	0	19	138	440	777	1194	1311	1156	992	564	239	45	6875	1	91
Binghamton (AP)	22	65	201	471	810	1184	1277	1154	1045	645	313	99	7286	- 2	91
Binghamton (PO)	0	28	141	406	732	1107	1190	1081	949	543	229	45	6451		
Buffalo	19	37	141	440	777	1156	1256	1145	1039	645	329	78	7062	- 5	90
Central Park	0	0	30	233	540	902	986	885	760	408	118	9	4871	11	94
JF Kennedy Intl.	0	0	36	248	564	933	1029	935	815	480	167	12	5219	17	91
LaGuardia	0	0	27	223	528	887	973	879	750	414	124	6	4811	12	93
Rochester	9	31	126	415	747	1125	1234	1123	1014	597	279	48	6748	2	91
Schenectady	0	22	123	422	756	1159	1283	1131	970	543	211	30	6650	- 5	90
Syracuse	6	28	132	415	744	1153	1271	1140	1004	570	248	45	6756	- 2	90
NC: Asheville	0	0	48	245	555	775	784	683	592	273	87	0	4042	13	91
Cape Hatteras	0	0	0	78	273	521	580	518	440	177	25	0	2612		
Charlotte	0	0	6	124	438	691	691	582	481	156	22	0	3191	18	96
Greensboro	0	0	33	192	513	778	784	672	552	234	47	0	3805	14	94
Raleigh	0	0	21	164	450	716	725	616	487	180	34	0	3393	16	95
Wilmington	0	0	0	74	291	521	546	462	357	96	0	0	2347	23	94
Winston Salem	0	0	21	171	483	747	753	652	524	207	37	0	3595	14	94
N. DAK: Bismarck	34	28	222	577	1083	1463	1708	1442	1203	645	329	117	8851	-24	95
Devils Lake	40	53	273	642	119[illegible]	[illegible]634	1872	1579	1345	753	381	138	9901	-23	93
Fargo	28	37	219	574	1107	1569	1789	1520	1262	690	332	99	9226	-22	92
Williston	31	43	261	601	1122	1513	1758	1473	1262	681	357	141	9243	-21	94

NORMAL TOTAL HEATING DEGREE DAYS (Base 65°)														Design-T_o °F	
STATE AND STATION	JULY	AUG	SEPT	OCT	NOV	DEC	JAN	FEB	MAR	APR	MAY	JUNE	ANNUAL	WIN.	SUMM.
OHIO: Akron	0	9	96	381	726	1070	1138	1016	871	489	202	39	6037	1	89
Cincinnati	0	0	54	248	612	921	970	837	701	336	118	9	4806	8	94
Cleveland	9	25	105	384	738	1088	1159	1047	918	552	260	66	6351	2	91
Columbus	0	6	84	347	714	1039	1088	949	809	426	171	27	5660	2	92
Dayton	0	6	78	310	696	1045	1097	955	809	429	167	30	5622	0	92
Mansfield	9	22	114	397	768	1110	1169	1042	924	543	245	60	6403	1	91
Sandusky	0	6	66	313	684	1032	1107	991	868	495	198	36	5796	4	92
Toledo	0	16	117	406	792	1138	1200	1056	924	543	242	60	6494	1	92
Youngstown	6	19	120	412	771	1104	1169	1047	921	540	248	60	6417	1	89
OKLA: Oklahoma City	0	0	15	164	498	766	868	664	527	189	34	0	3725	11	100
Tulsa	0	0	18	158	522	787	893	683	539	213	47	0	3860	12	102
OREG: Astoria	146	130	210	375	561	679	753	622	636	480	363	231	5186	27	79
Burns	12	37	210	515	867	1113	1246	988	856	570	366	177	6957		
Eugene	34	34	129	366	585	719	803	627	589	426	279	135	4726	22	91
Meacham	84	124	288	580	918	1091	1209	1005	983	726	527	339	7874		
Medford	0	0	78	372	678	871	918	697	642	432	242	78	5008	21	98
Pendleton	0	0	111	350	711	884	1017	773	617	396	205	63	5127	3	97
Portland	25	28	114	335	597	735	825	644	586	396	245	105	4635	26	91
Roseburg	22	16	105	329	567	713	766	608	570	405	267	123	4491	25	93
Salem	37	31	111	338	594	729	822	647	611	417	273	144	4754	21	92
Sexton Summit	81	81	171	443	666	874	958	809	818	609	465	279	6524		
PA: Allentown	0	0	90	353	693	1045	1116	1002	849	471	167	24	5810	3	92
Erie	0	25	102	391	714	1063	1169	1081	973	585	288	60	6451	7	88
Harrisburg	0	0	63	298	648	992	1045	907	766	396	124	12	5251	9	92
Philadelphia	0	0	60	291	621	964	1014	890	744	390	115	12	5101	11	93
Pittsburgh	0	9	105	375	726	1063	1119	1002	874	480	195	39	5987	7	90
Reading	0	0	54	257	597	939	1001	885	735	372	105	0	4945	6	92
Scranton	0	19	132	434	762	1104	1156	1028	893	498	195	33	6254	2	89
Williamsport	0	9	111	375	717	1073	1122	1002	856	468	177	24	5934	1	91
RI: Block Island	0	16	78	307	594	902	1020	955	877	612	344	99	5804		
Providence	0	16	96	372	660	1023	1110	988	868	534	236	51	5954	6	89

	NORMAL TOTAL HEATING DEGREE DAYS (Base 65°)													Design T, $^{\circ}$F	
STATE AND STATION	JULY	AUG	SEPT	OCT	NOV	DEC	JAN	FEB	MAR	APR	MAY	JUNE	ANNUAL	WIN.	SUMM.
SC: Charleston	0	0	0	59	282	471	487	389	291	54	0	0	2033	26	95
Columbia	0	0	0	84	345	577	570	470	357	81	0	0	2484	20	98
Florence	0	0	0	78	315	552	552	459	347	84	0	0	2387	21	96
Greenville	0	0	0	112	387	636	648	535	434	120	12	0	2884	19	95
Spartanburg	0	0	15	130	417	667	663	560	453	144	25	0	3074	18	95
S. DAK: Huron	9	12	165	508	1014	1432	1628	1355	1125	600	288	87	8223	-16	97
Rapid City	22	12	165	481	897	1172	1333	1145	1051	615	326	126	7345	- 9	96
Sioux Falls	19	25	168	462	972	1361	1544	1285	1082	573	270	78	7839	-14	95
TENN: Bristol	0	0	51	236	573	828	828	700	598	261	68	0	4143	11	92
Chattanooga	0	0	18	143	468	698	722	577	453	150	25	0	3254	15	97
Knoxville	0	0	30	171	489	725	732	613	493	198	43	0	3494	13	95
Memphis	0	0	18	130	447	698	729	585	456	147	22	0	3232	17	98
Nashville	0	0	30	158	495	732	778	644	512	189	40	0	3578	12	97
Oak Ridge (CO)	0	0	39	192	531	772	778	669	552	228	56	0	3817		
TEXAS: Abilene	0	0	0	99	366	586	642	470	347	114	0	0	2624	17	101
Amarillo	0	0	18	205	570	797	877	664	546	252	56	0	3985	8	98
Austin	0	0	0	31	225	388	468	325	223	51	0	0	1711	25	101
Brownsville	0	0	0	0	66	149	205	106	74	0	0	0	600	36	94
Corpus Christi	0	0	0	0	120	220	291	174	109	0	0	0	914	32	95
Dallas	0	0	0	62	321	524	601	440	319	90	6	0	2363	19	101
El Paso	0	0	0	84	414	648	685	445	319	105	0	0	2700	21	100
Fort Worth	0	0	0	65	324	536	614	448	319	99	0	0	2405	20	102
Galveston	0	0	0	0	138	270	350	258	189	30	0	0	1235	32	91
Houston	0	0	0	6	183	307	384	288	192	36	0	0	1396	29	96
Laredo	0	0	0	0	105	217	267	134	74	0	0	0	797	32	103
Lubbock	0	0	18	174	513	744	800	613	484	201	31	0	3578	11	99
Midland	0	0	0	87	381	592	651	468	322	90	0	0	2591	19	100
Port Arthur	0	0	0	22	207	329	384	274	192	39	0	0	1447	29	94
San Angelo	0	0	0	68	318	536	567	412	288	66	0	0	2255	20	101
San Antonio	0	0	0	31	207	363	428	286	195	39	0	0	1549	25	99
Victoria	0	0	0	6	150	270	344	230	152	21	0	0	1173	28	98
Waco	0	0	0	43	270	456	536	389	270	66	0	0	2030	21	101
Wichita Falls	0	0	0	99	381	632	698	518	378	120	6	0	2832	15	103

NORMAL TOTAL HEATING DEGREE DAYS (Base 65°)														Design T_o °F	
STATE AND STATION	JULY	AUG	SEPT	OCT	NOV	DEC	JAN	FEB	MAR	APR	MAY	JUNE	ANNUAL	WIN.	SUMM.
UTAH: Milford	0	0	99	443	867	1141	1252	988	822	519	279	87	6497		
Salt Lake City	0	0	81	419	849	1082	1172	910	763	459	233	84	6052	5	97
Wendoyer	0	0	48	372	822	1091	1178	902	729	408	177	51	4778		
VT: Burlington	28	65	207	539	891	1349	1513	1333	1187	714	353	90	8269	-12	88
VA: Cape Henry	0	0	0	112	360	645	694	633	536	246	53	0	3279		
Lynchburg	0	0	51	223	540	822	849	731	605	267	7[illegible]	0	4166	15	94
Norfolk	0	0	0	136	408	698	738	655	533	216	37	0	3421	20	94
Richmond	0	0	36	214	495	784	815	703	546	219	53	0	3865	14	96
Roanoke	0	0	51	229	549	825	834	722	614	261	65	0	4150	15	94
Wash. Nat'l AP	0	0	33	217	519	834	871	762	626	288	74	0	4224		
WASH: Olympia	68	71	198	422	636	753	834	675	645	450	307	177	5236	21	85
Seattle	50	47	129	329	543	657	738	599	577	396	242	177	4424	23	82
Seattle Boeing	34	40	147	384	624	763	831	655	608	411	242	99	4838		
Seattle Tacoma	56	62	162	391	633	750	828	678	657	474	295	159	5145	20	85
Spokane	9	25	168	493	879	1082	1231	980	834	531	288	135	6655	- 2	93
Stampede Pass	273	291	393	701	1008	1178	1287	1075	1085	855	654	483	9283		
Tatoosh Island	295	279	306	406	534	639	713	613	645	525	431	333	5719		
Walla Walla	0	0	87	310	681	843	986	745	589	342	177	45	4805	12	98
Yakima	0	12	144	450	828	1039	1163	868	713	435	220	69	5941	6	94
W. VA: Charleston	0	0	63	254	591	865	880	770	648	330	96	9	4476	9	92
Elkins	9	25	135	400	729	992	1008	896	791	444	198	48	5675	1	87
Huntington	0	0	63	257	585	856	880	764	636	294	99	12	4446	10	95
Parkersburg	0	0	60	264	606	905	942	826	691	339	115	6	4754	8	93
WIS: Green Bay	28	50	174	484	924	1333	1494	1313	1141	654	335	99	8029	-12	88
La Crosse	12	19	153	437	924	1339	1504	1277	1070	540	245	69	7589	-12	90
Madison	25	40	174	474	930	1330	1473	1274	1113	618	310	102	7863	- 9	92
Milwaukee	43	47	174	471	876	1252	1376	1193	1054	642	372	135	7635	- 6	90
WYO: Casper	6	16	192	524	942	1169	1290	1084	1020	657	381	129	7410	-11	92
Cheyenne	19	31	210	543	924	1101	1228	1056	1011	672	381	102	7278	-6	89
Lander	6	19	204	555	1020	1299	1417	1145	1017	654	381	153	7870	-16	92
Sheridan	25	31	219	538	948	1200	1355	1154	1054	642	366	150	7683	-12	95

TRAINING COURSE IN
THE PRACTICAL ASPECTS OF
DESIGN OF SOLAR HEATING AND COOLING SYSTEMS
FOR
RESIDENTIAL BUILDINGS

MODULE 5

HEATING AND COOLING LOAD ANALYSES

SOLAR ENERGY APPLICATIONS LABORATORY
COLORADO STATE UNIVERSITY
FORT COLLINS, COLORADO

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GLOSSARY OF TERMS

Cooling Load	Rate of heat removal from a building to maintain constant indoor temperature.
Conductance	Expresses the ease or difficulty of materials to conduct heat from the high temperature side to the low temperature side.
Design Equivalent Temperature Differential	Temperature differential between outdoor and indoor temperature adjusted for effects of surface absorptance of solar energy and radiation.
Heat Gains	Rate of heat flow into a building per unit time, usually one hour.
Heat Loss	Rate of heat flow out of a building per unit time, usually one hour.
Heat Transmission Losses	Heat loss
Infiltration Gain	Infiltration of hot air into a building which must be cooled to the comfort level of the building air.
Infiltration Loss	Infiltration of cold air into a building which must be heated to the comfort level of the building air.
R value	Thermal resistance of materials to flow of heat.
U value	Heat transfer coefficient for material.

LIST OF SYMBOLS

A	Area of heat transfer surface, ft^2
f_i	Air film conductance for still air on inside surface of the building, Btu/(hr)(ft^2)(^{0}F)
f_o	Air film conductance for moving air on outside surface of the building, Btu/(hr)(ft^2)(^{0}F)
Q	Heat flow rate, Btu/hr
R	Thermal resistance of building materials, (hr)(ft^2)(^{0}F)/Btu
R_i	Thermal resistance of inside air film, (hr)(ft^2)(^{0}F)/Btu
R_o	Thermal resistance of outside air film, (hr)(ft^2)(^{0}F)/Btu
R_t	Total thermal resistance for composite building components, (hr)(ft^2)(^{0}F)/Btu
T_i	Design inside temperature, ^{0}F
T_G	Design garage temperature, ^{0}F
T_o	Design outdoor temperature, ^{0}F
U	Heat transfer coefficient, Btu/(hr)(ft^2)(^{0}F)
U_o	Overall heat transfer coefficient, Btu/(hr)(ft^2)(^{0}F)
V	Volume change of air in the rooms per hour, ft^3/hr
W_i	Humidity ratio of indoor air, dimensionless
W_o	Humidity ratio of outdoor air, dimensionless

INTRODUCTION

TRAINEE-ORIENTED OBJECTIVE

The objective of this module is to present methods for estimating heat loss and heat gains for a building to be heated and cooled, based on maintaining a selected indoor air temperature during periods of design outdoor weather conditions.

SUB-OBJECTIVES

The trainee will be able to:

1. Select the design outdoor temperature
2. Determine heat transfer coefficients
3. Compute heat transmission losses
4. Compute the design heat loss for the building based on degree-days
5. Compute heat gains in the building
6. Compute the design cooling load for the building.

Proper design of space heating and air conditioning requires knowledge of heat losses and gains for a building. This is particularly needed for solar systems because the thermal loads affect system size, and system size affects the costs.

This module is concerned with the procedures for determining the heat transmission losses and gains and the necessary data to perform the calculations. Various methods for determining thermal loads for buildings are available and the trainees are undoubtedly familiar with some of them. Only one method for determining heat losses and one for heat gains is presented herein to establish a common base for the trainees in this course.

This is not to imply that other methods are not useful. The methods described in this module are simple to apply.

HEAT LOSSES

Heat transmission losses, or more simply heat losses, from buildings may be divided into two groups: (1) the transmission losses through walls, floor, ceiling, glass and other surfaces and (2) the infiltration losses, or more correctly infiltration of cold air; through open doors and windows, cracks and crevices around doors and windows, which must be heated to the comfort level in the building.

HEAT TRANSMISSION THROUGH BUILDING SURFACES

Heat is transferred from warm room air to outdoor air by a three-step process. Heat is transferred from the room air to the inside surface of a wall or window, through the wall or window, and from the outside surface to the outdoor air. The rate of heat flow per unit time from the building to the outdoors depends upon the surface area, A, an overall heat transfer coefficient, U, and the air temperature difference between the inside, T_i, and outside, T_o. Expressed in equation form:

$$Q = UA(T_i - T_o) \tag{5-1}$$

where Q is heat flow rate, Btu/hr; A is wall area, ft^2; U is the overall heat transfer coefficient, Btu per $(hr)(ft^2)(^oF)$; T_i is indoor temperature, ^oF; and T_o is outdoor temperature, ^oF.

The overall heat transfer coefficient, often called the U factor, is determined by the reciprocal of the total thermal resistance, R_T, to heat flow:

$$U = \frac{1}{R_T} \qquad (5\text{-}2)$$

and

$$R_T = R_1 + R_2 + R_3 + R_4 + \text{ etc.} \qquad (5\text{-}3)$$

where R_1, R_2, etc., are R factors, the individual resistances ~~of the~~ wall components.

The transfer of heat from the inside air to the wall is visualized as taking place through a thin film of air adjacent to the wall surface. This thin film has resistance, R_i, to heat flow determined by the film conductance, f_i,

$$R_i = \frac{1}{f_i} \qquad (5\text{-}4)$$

and should be included in the determination of the overall U factor. Similarly, there is a thin film at the outside surface, the conductance of which, symbolized by f_o, is dependent upon the wind speed. The resistance of the outside film, R_o, is

$$R_o = \frac{1}{f_o} \qquad (5\text{-}5)$$

During summer months, when the outside temperature is greater than the indoor temperature, heat is conducted into the building. The principles are the same as the foregoing, except that heat flow rate is determined by

$$Q = UA\ (T_o - T_i) \qquad (5\text{-}6)$$

where T_o and T_i have been interchanged from equation (5-1).

Surface conductances and resistances for air films for interior and exterior surfaces, for winter and summer, are tabulated in Table 5-1 at the end of this module. The winter values are based on wind velocity of 15 mph and summer values are based on wind velocity of 7 mph.

Dead air spaces between walls offer thermal resistance. The resistance values are tabulated in Table 5-2 for 3/4-inch and 4-inch spaces for winter and summer conditions. For spaces between 3/4 and 4 inches, values may be interpolated.

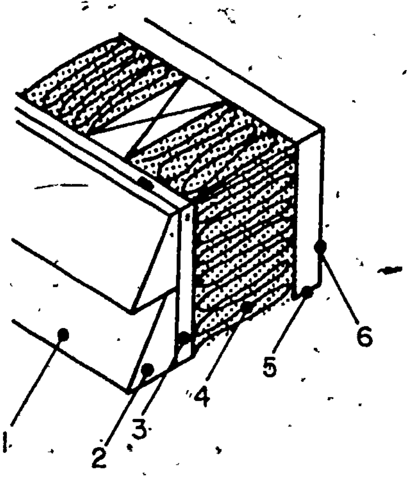
Resistance values for common building materials are tabulated in Table 5-3. U factors for windows and patio doors are tabulated in Table 5-4, and U factors for solid doors are listed in Table 5-5 with and without storm doors. The values in these tables correspond with more complete tables listed in Chapter 20, ASHRAE Handbook of Fundamentals (1972).

TRANSMISSION COEFFICIENTS

The procedure for determining the overall heat transmission coefficients, U, for typical wall, roof, ceiling and floor construction is presented in this section. The values of R used are found in Tables 5-1 through 5-3. U factors for composite construction are determined in the following examples. U factors for other types of construction may be calculated by following these examples.

Example 1 - Frame Wall (2 x 4 studs)

	ITEM	R
1.	Outside film (15 mph wind, winter)	0.17
2.	Siding, wood (½ x 8 lapped)	0.81
3.	Sheathing (½ inch regular)	1.32
4.	Insulation batt (3-3½ inch)	11.00
5.	Gypsum wall board (½ inch)	0.45
6.	Inside surface (winter)	0.68
	Total Resistance, R_T	14.43
	$U = 1/R_T$	0.07



The calculated U factor applies to the area between 2 x 4 studs. Because the resistance to heat flow through the 2 x 4 stud is different from the insulation, a correction is sometimes applied. However, the corrections usually amount to less than the accuracy of the R values. Corrections are therefore considered unnecessary.

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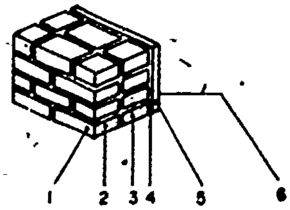
Example 2 - Frame Wall (2 x 6 studs)

From Example 1, R_T	14.43
Replace 3½-inch insulation, subtract	11.00
	3.43
Add 5½-inch insulation	19.00
New R_T	22.43
$U = 1/R_T$	0.04
Difference in U from Example 1	0.03
Percent Difference from 2 x 4 wall	43 percent

There is 43-percent reduction ~~in~~ heat loss for a 2 x 6 wall as compared with a 2 x 4 wall with correspondingly thicker insulation in the 2 x 6 wall.

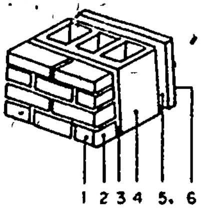
Example 3 - Solid Masonry Wall

	ITEM	R
1.	Outside film (15 mph wind, winter)	0.17
2.	Face brick (4 inch)	0.44
3.	Common brick (4 inch)	0.80
4.	Air space (3/4 inch)	1.28
5.	Gypsum board (½ inch)	0.45
6.	Inside surface	0.68
	Total Resistance, R_T	3.82
	$U = 1/R_T$	0.26



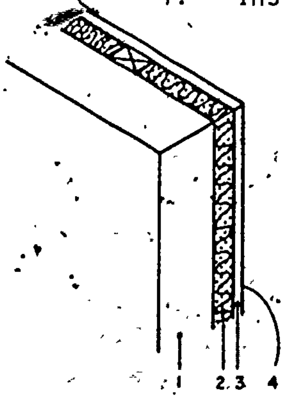
Example 4 - Masonry Walls

	ITEM	R
1.	Outside surface (15 mph)	0.17
2.	Face brick (4 inch)	0.44
3.	Cement mortar (½ inch)	0.10
4.	Cinder block (8 inch)	1.72
5.	Air space (3/4 inch)	1.28
6.	Gypsum board (½ inch)	0.45
7.	Inside surface	0.68
	Total Resistance, R_T	4.84
	$U = 1/R_T$	0.21



Example 5 - Basement Wall

	ITEM	R
1.	Concrete wall (8 inch)	0.64
2.	Insulation batt (2 inch)	7.00
3.	Gypsum board (½ inch)	0.45
4.	Inside surface	0.68
	Total Resistance, R_T	8.77
	$U = 1/R_T$	0.11



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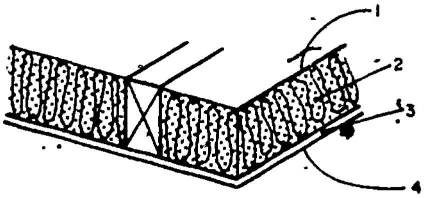
Example 6 - Insulated Ceiling, 6 inches

	ITEM	R
1.	Inside surface	0.68
2.	Insulation batt (6 inch)	19.00
3.	Gypsum board (½ inch)	0.45
4.	Inside surface	0.68
	Total Resistance, R_T	20.81
	$U = 1/R_T$	0.05



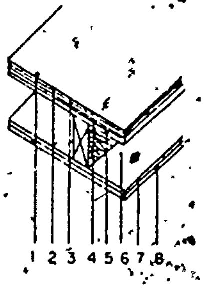
Example 7 - Insulated Ceiling, 9 inches

	ITEM	R
1.	Inside surface	0.61
2.	Insulation (9 inch)	24.00
3.	Gypsum board (½ inch)	0.45
4.	Inside surface	0.61
	Total Resistance, R_T	25.67
	$U = 1/R_T$	0.04

Percent decrease of U with 9-inch insulation over 6-inch insulation, 20 percent.

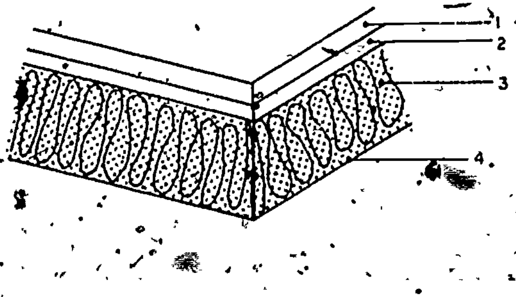
Example 8 - Floor

	ITEM	R
1.	Top surface	0.61
2.	Linoleum or tile	0.05
3.	Felt	0.06
4.	Plywood (5/8 inch)	0.78
5.	Wood subfloor (3/4 inch)	0.94
6.	Air space	0.85
7.	Acoustic ceiling tile (3/4 inch)	1.89
8.	Surface	0.61
	Total Resistance, R_T	5.79
	$U = 1/R_T$	0.17



Example 9 - Floor

	ITEM	R
1.	Carpet and fibrous pad	2.08
2.	Plywood (3/4 inch)	0.93
3.	Insulation (9 inch)	24.00
4.	Surface (still air)	0.61
	Total Resistance, R_T	27.62
	$U = 1/R_T$	0.04

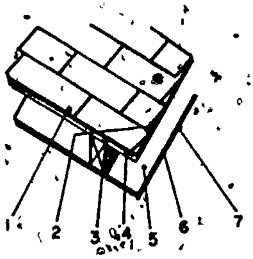


Example 10 - Basement

The heat loss from a heated basement should be based on a heat transfer coefficient for both wall and floor of $U = 0.10$. The temperature adjacent to basement walls and floor varies with the rate of heat transfer through the walls. The more heat that flows through the walls, the warmer will be the ground temperature. Below basement floors, a ground temperature equal to the ground water temperature is sometimes used. A temperature of 45°F is recommended as a rule of thumb for this course. If conditions warrant, a different temperature may be used.

Example 11 - Pitched Roofs (Heat Flow Up)

	ITEM	R
1.	Outside surface (15 mph)	0.17
2.	Asphalt shingle roofing	0.44
3.	Building paper	0.06
4.	Plywood deck (5/8 inch)	0.78
5.	Inside surface	0.61
	Total Resistance, R_T	2.06
	$U = 1/R_T$	0.49



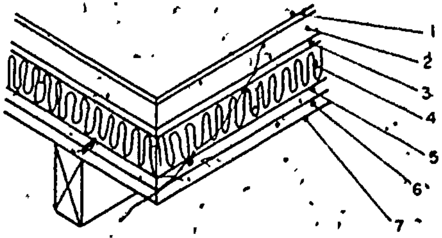
Example 12 - Pitched Roof with Air Space and Sheathing (Heat Flow Up)

See Example 11.

	ITEM	R
1.	Outside surface	0.17
2.	Wood shingle	0.94
3.	15 pound felt	0.06
4.	Plywood deck (5/8 inch)	0.78
5.	Air space	1.00
6.	Gypsum (½ inch)	0.45
7.	Inside surface	0.61
	Total Resistance, R_T	4.01
	$U = 1/R_T$	0.25

Example 13 - Pitched Roof with Mounted Collector (At Night)

	ITEM	R
1.	Outside surface	0.17
2.	Glass	1.13
3.	Air space (3/4 inch)	1.75
4.	Insulation	7.00
5.	15 pound felt	0.06
6.	Plywood (3/4 inch)	0.93
7.	Inside surface	0.61
	Total Resistance, R_T	11.65
	$U = 1/R_T$	0.09



HEAT LOSS BY INFILTRATION

Calculation of infiltration losses can be very complex. Experience and judgment are important to provide reasonable estimates. Of two methods used for calculating infiltration rates, only the simpler air change method is discussed in this module. Readers are referred to the ASHRAE Handbook of Fundamentals for details of the "Crack" method. In either method, the objective is to determine the amount of heat required to raise the temperature of cold air which enters a building through cracks, open windows, and doors.

The volume of cold air expected to enter a room through cracks during a one-hour period depends on such factors as wind direction and speed, pressure differences inside and outside the building, storm windows, air locks on outdoor entrances, and whether room doors are closed. The entering volume of cold air is expressed in terms of air changes per hour in the room under consideration. It is normally expected that storm doors and windows, or tight-fitting double-glazed windows will soon be widely adopted in new construction, particularly for solar heated and cooled houses. The average air changes for rooms with various fenestrations listed in Table 5-6 are in accordance with Chapter 19, ASHRAE Handbook of Fundamentals (1972).

From the air change rate, per hour, the volume rate of air change per hour, V, is determined from the room volume. The heat loss from infiltration is calculated from

$$Q = 0.018\ V\ (T_i - T_o) \qquad (5\text{-}7)$$

where V is the volume change per hour; Q is Btu per hour.

When moisture is added to the air to maintain winter comfort conditions, heat will be required to evaporate the water vapor added to the building air. The rate of heat added is most conveniently calculated from the equation below:

$$Q = 79.5\ V \cdot (W_i - W_o) \qquad (5\text{-}8)$$

where V is the infiltration rate, cfh; W_i is humidity ratio of indoor air, dimensionless; W_o is humidity ratio of outdoor air, dimensionless.

Infiltration occurs primarily because of wind impacting on the building from a given direction. Therefore, only the rooms on one side of the building would be affected at a given time. The values in Table 5-6 account for this factor.

HEAT LOSS CALCULATION

Procedure

1. Select the design outdoor temperature, T_o, for selected cities from the last column in Table 4-5.
2. Select the indoor design temperature, T_i, at 68 °F. (If zone controls or clock thermostats are used to lower the temperature of unused rooms and at night, consideration should be given to selecting other indoor temperatures for specific periods of time.)
3. Determine net areas, A, of walls, roof, ceiling, windows, doors, and floor for each different type of construction.
4. Select U factors from Examples 1 through 13, or calculate appropriate U factors for specific wall type.
5. Calculate heat transmission loss rate from:

   $$Q = UA(T_i - T_o) \qquad (5\text{-}1)^*$$

   through each type of surface.

---

[*] See section on temperatures of unheated spaces.

6. Sum the transmission losses.
7. Determine infiltration losses.
8. Add the infiltration losses to the transmission losses to obtain the total heat loss from the building.
9. Determine the design heat loss rate for the building for each degree day.

Temperatures of Unheated Spaces

Attic Temperature -- The attic temperature is determined from a balance of heat flow into and out of the attic. Heat flow into the attic is from the ceiling; heat flow out is through the roof surfaces and end walls. The general formula for determining attic temperature is:

$$T_{at} = \frac{A_c U_c T_c + T_o(A_r U_r + A_w U_w)}{A_c U_c + A_r U_r + A_w U_w} \qquad (5\text{-}9)$$

where

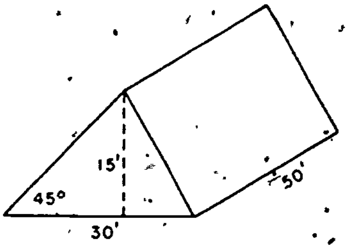
T_{at} is attic temperature, ^{o}F
T_c is room temperature, ^{o}F
T_o is outside temperature, ^{o}F
A_c is ceiling area, ft^2
A_r is roof area, ft^2
A_w is roof wall area, ft^2
U_c is ceiling U factor, $Btu/(hr)(ft^2)(^{o}F)$
U_r is roof U factor, $Btu/(hr)(ft^2)(^{o}F)$
U_w is wall U factor, $Btu/(hr)(ft^2)(^{o}F)$

Example 14 - Attic Temperature for a Wood Shingled Roof

Calculate attic temperature for a wood shingled roof with the given dimensions. T_o is -9 ^oF, T_c is 68 ^oF. See Example 6 for ceiling U factor, U_c = 0.05. See Example 12 for roof U factor, U_r = 0.25. For Example 1, for no insulation and 3½-inch air space, U factor for wall is:

R_T from Example 1	14.43
Subtract insulation	-11.00
Subtract gypsum board	- 0.45
Total Resistance, R_T	2.98
$U_w = 1/R_T$	0.34

Calculate Area:



A_c = 30 x 50 = 1500 ft^2

A_r = $\sqrt{2}$ x 15 x 50 x 2 = 2120 ft^2

A_w = 30 x 15 x ½ x 2 = 450

$$T_{at} = \frac{(1500)(0.05)(68) + (-9)\ [(2120)(0.25) + (450)(0.34)]}{(1500)(0.5) + (2120)(.25) + (450)(0.34)}$$

$$T_{at} = \frac{5100 - 6147}{75 + 530 + 153} = -1.4\ ^oF$$

Example 15 - Attic Temperature with Mounted Collector

Calculate the attic temperature with a collector mounted on one side of roof. From Example 13, U_r with collector is 0.09. A_rU_r in equation (5-9) consists of two parts:

A_r (with collector) = 1060 ft^2

A_r (without collector) = 1060 ft^2

U_r (with collector) = 0.09

U_r (without collector) = 0.25

$$A_rU_r = (1060)(0.9) + (1060)(.25) = 360$$

For Example 5-14,

$$T_{at} = \frac{(1500)(0.5)(68) + (-9)[360 + (450)(.34)]}{(1500)(0.5) + 360 + 153}$$

$$T_{at} = \frac{5100 - 4617}{588} = 0.8\ ^oF$$

When ventilation is provided, at 0.5 cfm per square foot of ceiling, the attic temperatures must be reduced from those calculated in Examples 14 and 15. Thus, the attic temperature approaches outdoor temperature. Attic temperature may be assumed to be the outdoor temperature with well-insulated ceilings without significant error in heat loss calculation.

Unheated Garage. -- With similar detailed calculations, the temperature in any unheated garage may be calculated. For ease of calculation of heat losses, the garage temperature may be assumed to be the mean of the indoor and outdoor temperatures, thus:

$$T_G = \frac{T_o + T_i}{2} \qquad (5\text{-}10)$$

Example: With outdoor temperature of -9 ^oF, indoor temperature of 68 ^oF, the garage temperature is:

$$T_G = \frac{(-9) + 68}{2} = 30\ ^oF$$

Example Heat Loss Calculation

An example heat loss calculation is presented below for a house in Fort Collins shown in Figure 5-1, with the description of materials given in Figure 5-2. The windows in all bedrooms are 3' x 4', double hung, single pane, wood sash with storm windows having 3-inch air space. The window in the bathroom is 2'x 2', double hung, single pane, wood sash with storm window. The window in the living room is 4' x 8', wood sash, double glass with ½-inch air space. The window in the kitchen is 2.5' x 4' double hung, single pane, wood sash with storm window. The window in the breakfast nook is 3' x 4' double glass, wood sash with ½-inch air space. The 6' x 6' sliding patio door in the family room is double-glass wood frame with ½-inch air space. The basement windows are 1½' x 1½' and will be ignored in this calculation. Bathrooms and kitchen are ventilated.

The heating worksheet in Figure 5-3 is used to facilitate calculations. Referring to Table 4-5, the design temperature is -9 °F for Fort Collins, Colorado. The design indoor temperature is 68 °F. The total heat loss from the building for the design temperatures is 53,215 Btu per hour. The heat load based on degree days is determined as follows:

$$\frac{\text{Heat loss}}{\text{Design Temperature Diff.}} \times 24$$

For the example of Figure 5-3, the heat load based on degree days (DD) is [68 - (-9)]

$$\frac{53,215 \times 24}{68 - (-9)} = 16590 \ \frac{\text{Btu}}{\text{DD}}$$

It is interesting to note that the overall U factor for the house for the above grade living space based on the computations in Figure 5-3 is

$$U_o = \frac{(53215 - 7921)}{2078 \times 77} = 0.28 \text{ Btu/(hr)(ft}^2\text{)(}^\circ\text{F)}$$

For the entire house including the basement,

$$U_o = \frac{53,215}{3260 \times 77} = 0.21 \text{ Btu/(hr)(ft}^2\text{)(}^o\text{F)}$$

HEAT GAINS

Heat transmission into a building takes place by radiation and conduction from building surfaces and by infiltration of warm air into conditioned space. The detailed procedure is quite complex, taking into account the thermal and optical properties of the building materials, time of day, day of the year, solar radiation intensity, etc. The procedure described in this module is based on a simplified method using a design equivalent temperature difference.

Heat gain is computed by:

$$Q = UA(DTD) \qquad (5\text{-}11)$$

where

- Q is rate of heat gain, Btu/hr
- A is area of surface, ft^2
- U is heat transmission coefficient, $Btu/(hr)(ft^2)(^oF)$
- DTD is design equivalent temperature difference.

The DTD for three design outdoor temperatures are listed in Table 5-7. U factors for typical construction may be computed in the manner shown in Examples 1 through 13. Heat gain through windows depends upon exposure to solar radiation; therefore, heat gains will differ for different window orientations. Heat gains directly in terms of $Btu/(hr)(ft^2)$ are listed in Table 5-8. No credit is given for shade line below an overhang in the table. When a permanent overhang is provided, the shaded window is treated as a north-facing window. Average shade lines below an overhang for various latitudes and window orientation are given in Table 5-9. The overhang width multiplied by the shade factor determines the average effective shadow lines below the level of the overhang. Data are for August 1, averaged over 5 hours.

INFILTRATION

Infiltration in the summer is less than in winter because the temperature difference and wind velocity are less. Air changes per hour for the summer are listed in Table 5-6. Sensible heat gain is determined by equation (5-7) and latent heat gain by equation (5-8). Residential cooling loads are almost always based on sensible heat gains.

OCCUPANCY

Heat gain from human beings in a residence is usually assumed to be about 200 to 250 Btu per hour. For normally equipped kitchens, heat gain from appliances is assumed to be 1200 Btu per hour for determining cooling loads.

SOLAR EQUIPMENT

Heat gains from solar equipment in a residence, i.e., motors, heated pipes and ducts, will add to the cooling load. The heat gain could be significant from water storage tanks if the equipment room is not vented. While there are as yet insufficient data from solar heated and cooled houses to provide design tables, a heat gain equivalent to the kitchen load, 1200 Btuh, may be assumed.

LATENT HEAT

Latent heat load of 30 percent of the sensible heat load may be used.

COOLING LOAD

The differences between heat gains and cooling loads are important in calculating residential cooling loads. The cooling loads in residential buildings are primarily due to sensible heat flow and not to internal heat gains. It must be remembered that only a few days each season are design days, and a partial load condition exists for many hours during a season. Thus, an oversized system does not perform effectively with short term or intermittent operating cycles. Equipment should be of the smallest possible capacity and designed to operate for 24 hours a day, using the thermal storage available in interior walls, and furnishings, to reduce temperature excursions in the building.

PROCEDURE FOR CALCULATION

1. Determine the design outdoor summer temperature from Table 4-5.
2. Establish an indoor design temperature (usually 75 °F).
3. Determine net areas of building sub-structures.
4. Select U factors from Examples 1 through 13, or calculate U-factor from appropriate tables.
5. Select the Design Equivalent Temperature Difference (DETD) from Table 5-7.
6. For windows, use heat gain rates given in Table 5-8 corrected for shading factors given in Table 5-9.

7. Calculate the sensible heat gain from conduction and radiation, using equation (5-11).

$$Q = UA(DETD) \qquad (5\text{-}11)$$

8. Calculate the sensible heat gain due to infiltration, using Table 5-6.
9. Add heat gain from occupants and fixed appliances.
10. Sum the sensible heat gains.
11. Add 30 percent for latent cooling load.
12. Total the latent load and sensible heat gains to determine the total cooling load.

EXAMPLE

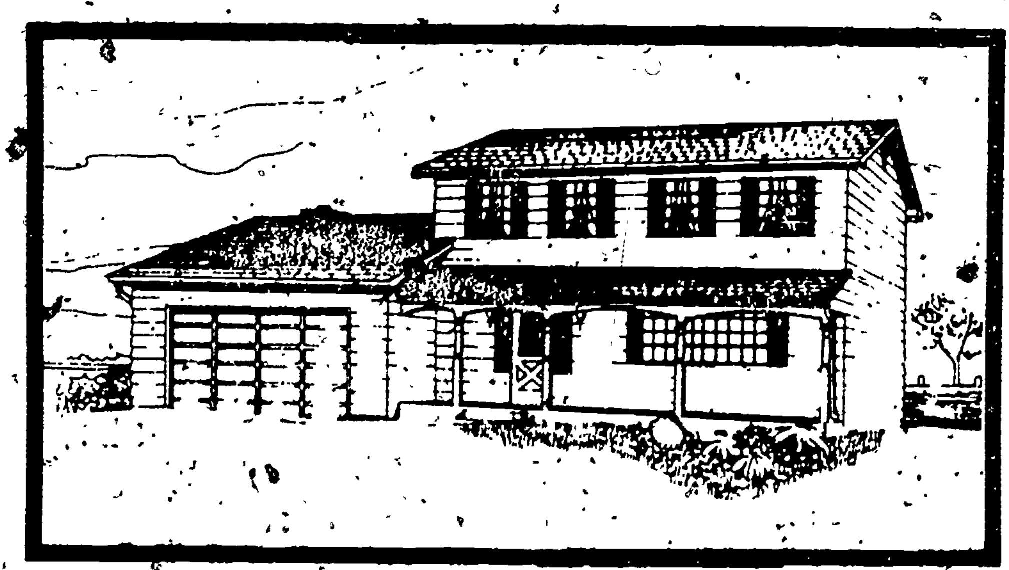
The cooling load for the house of Figure 5-1 is calculated as shown in Figure 5-4. The outdoor design temperature, from Table 4-5, is 89 ^{o}F. The indoor design temperature is 75 ^{o}F. The U factors for walls, ceiling and door are the same as for winter conditions. Refinement in U factors were not made in these computations although the R factors in air films in Tables 5-1 and 5-2 would result in slightly different R factors.

The overhangs over the south-facing windows effectively reduce the heat transfer rates equivalent to the north-facing windows, and there are no east and south-facing windows. No credit was taken for shades or drapes over the windows.

The temperature in the garage was assumed to be the mean between indoor and outdoor design temperatures, and the design temperature differences (DETD) given in Table 5-7 were interpolated for the design outdoor temperature of 89 ^{o}F.

The total cooling load for the building is calculated to be 18,621 Btu per hour. This low cooling load is a result of low design outdoor temperature in Fort Collins, 89 °F, and a building which is insulated properly with shading over windows. The values used apply for average summer conditions, and it is likely that cooling loads for days when temperatures reach 95 °F will require greater cooling capacity. If the air conditioner is operating 24 hours per day, even for these days, the temperature excursion inside the building should not be large.

Based on a cooling load of 18,621 Btu/hr, a temperature difference of 14 °F and above grade floor area, the overall heat transfer coefficient for the building is 0.64 Btu per hour per square foot of floor space for each °F temperature difference between design outdoor and indoor temperatures.

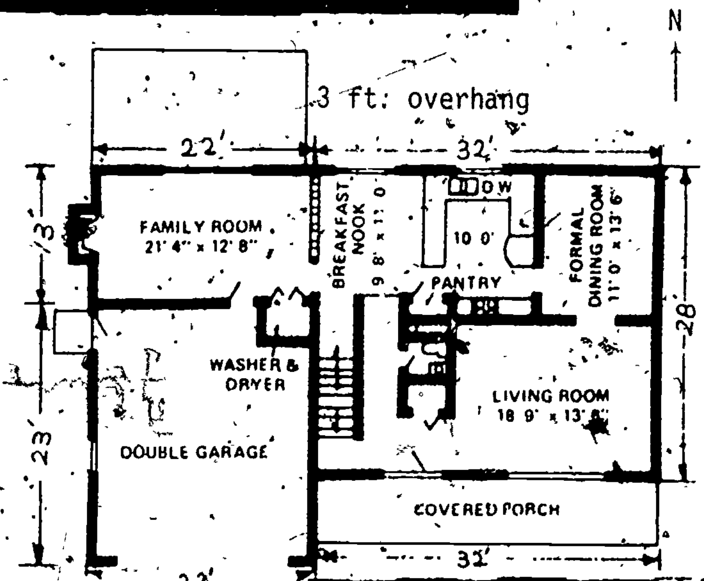


TOTAL FLOOR AREA

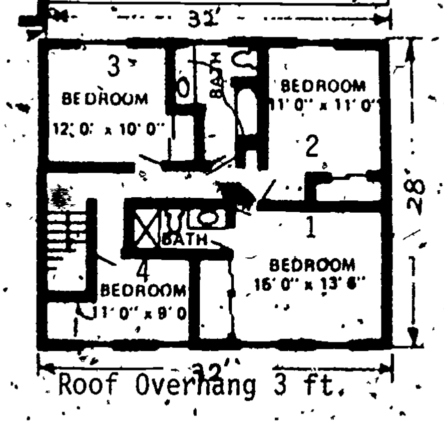
2078 ft^2 - 2 floors

plus 1182 ft^2 - Base.

3260 ft^2


N
3 ft. overhang
22'
32'
13
23
28
FAMILY ROOM 21' 4" x 12' 8"
BREAKFAST NOOK 9' 8" x 11' 0"
DW
10 0'
PANTRY
FORMAL DINING ROOM 11' 0" x 13' 6"
WASHER & DRYER
LIVING ROOM 18' 9" x 13' 6"
DOUBLE GARAGE
COVERED PORCH
22'

Colonial two-story with all the necessary size and luxury for a large or growing family • Four Bedrooms and Two Baths on the Second Floor • Large Entry With Open Stairway • Spacious Living Room • Formal Dining Room • U-Shape Kitchen With Eating Space • Family Room With Fireplace Located Next To Kitchen • Full Unfinished Basement • Two Car Garage • Paneling.


32'
3
BEDROOM 12' 0" x 10' 0"
BATH
BEDROOM 11' 0" x 11' 0"
2
1
BATH
BEDROOM 16' 0" x 13' 6"
4
BEDROOM 11' 0" x 9' 0"
28'
32'
Roof Overhang 3 ft.

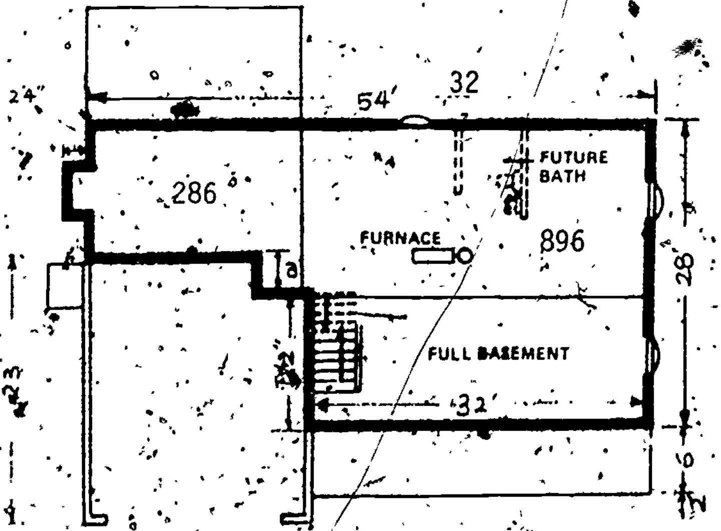
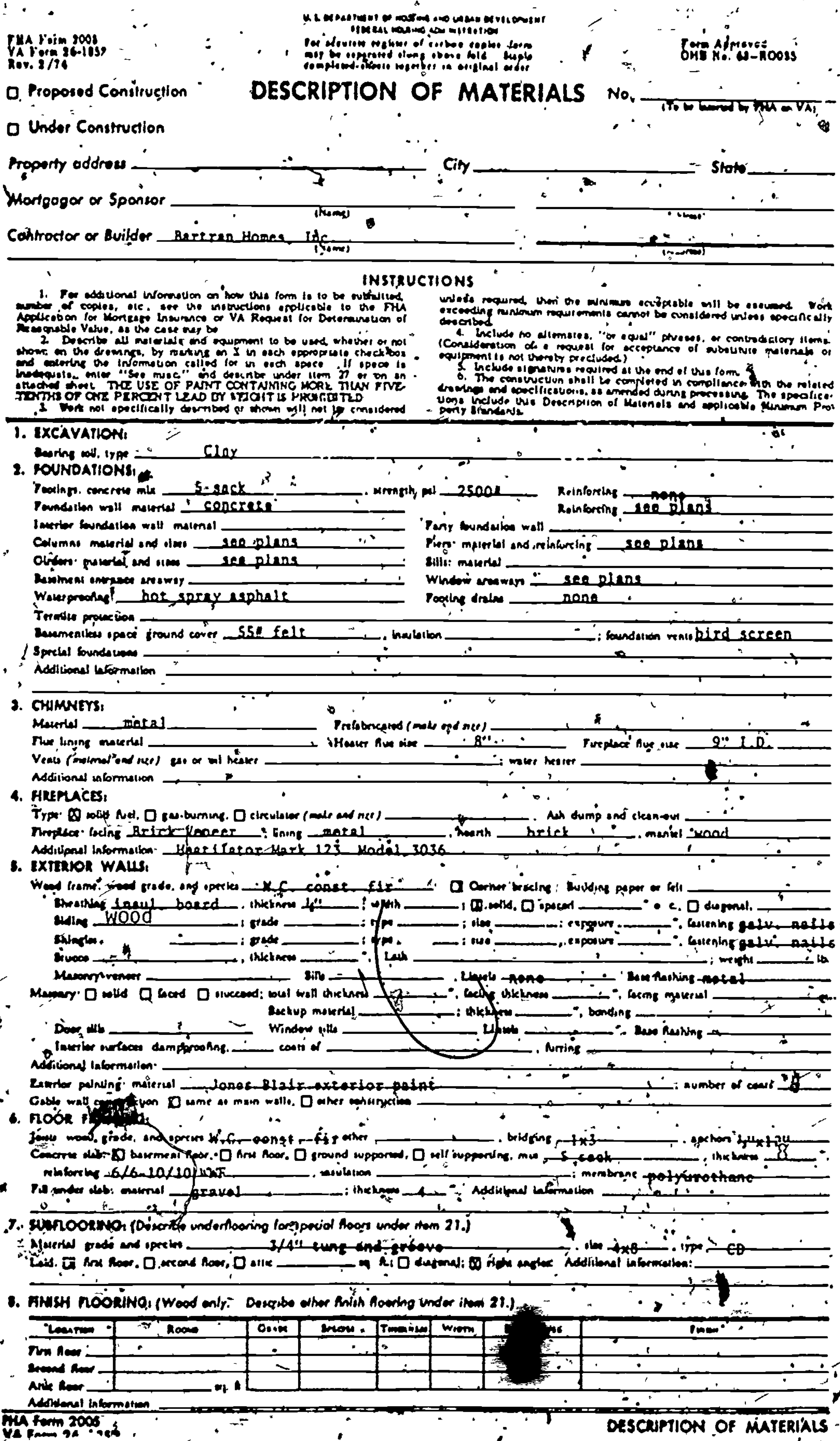

24"
54'
32
286
FUTURE BATH
FURNACE
896
28
a
FULL BASEMENT
32'
23
22
10

Figure 5-1. Example Residential Building

Figure 5-2



FHA Form 2005
VA Form 26-1852
Rev. 3/74

U.S. DEPARTMENT OF HOUSING AND URBAN DEVELOPMENT
FEDERAL HOUSING ADMINISTRATION
For accurate register of carbon copies form may be separated along above fold. Staple completed sheets together in original order

Form Approved
OMB No. 63-R0055

☐ Proposed Construction

☐ Under Construction

DESCRIPTION OF MATERIALS

No. ______ (To be inserted by FHA or VA)

Property address ______ City ______ State ______

Mortgagor or Sponsor ______ (Name) ______ (Address)

Contractor or Builder Bartran Homes Inc. (Name) ______ (Address)

INSTRUCTIONS

1. For additional information on how this form is to be submitted, number of copies, etc., see the instructions applicable to the FHA Application for Mortgage Insurance or VA Request for Determination of Reasonable Value, as the case may be.

2. Describe all materials and equipment to be used, whether or not shown on the drawings, by marking an X in each appropriate check-box and entering the information called for in each space. If space is inadequate, enter "See misc." and describe under item 27 or on an attached sheet. THE USE OF PAINT CONTAINING MORE THAN FIVE-TENTHS OF ONE PERCENT LEAD BY WEIGHT IS PROHIBITED.

3. Work not specifically described or shown will not be considered unless required, then the minimum acceptable will be assumed. Work exceeding minimum requirements cannot be considered unless specifically described.

4. Include no alternates, "or equal" phrases, or contradictory items. (Consideration of a request for acceptance of substitute materials or equipment is not thereby precluded.)

5. Include signatures required at the end of this form.

6. The construction shall be completed in compliance with the related drawings and specifications, as amended during processing. The specifications include this Description of Materials and applicable Minimum Property Standards.

1. EXCAVATION:

Bearing soil, type Clay

2. FOUNDATIONS:

Footings: concrete mix 5-sack; strength psi 2500# Reinforcing none

Foundation wall: material concrete Reinforcing see plans

Interior foundation wall: material ______ Party foundation wall ______

Columns: material and sizes see plans Piers: material and reinforcing see plans

Girders: material and sizes see plans Sills: material ______

Basement entrance areaway ______ Window areaways see plans

Waterproofing hot spray asphalt Footing drains none

Termite protection ______

Basementless space: ground cover 55# felt; insulation ______; foundation vents bird screen

Special foundations ______

Additional information ______

3. CHIMNEYS:

Material metal Prefabricated (make and size) ______

Flue lining: material ______ Heater flue size 8" Fireplace flue size 9" I.D.

Vents (material and size): gas or oil heater ______; water heater ______

Additional information ______

4. FIREPLACES:

Type: ☒ solid fuel; ☐ gas-burning; ☐ circulator (make and size) ______ Ash dump and clean-out ______

Fireplace: facing Brick Veneer; lining metal; hearth brick; mantel wood

Additional information: Heatilator Mark 123 Model 3036

5. EXTERIOR WALLS:

Wood frame: wood grade, and species W.C. const. fir ☐ Corner bracing. Building paper or felt ______

Sheathing insul. board; thickness 1/2"; width ______; ☒ solid; ☐ spaced ______" o. c.; ☐ diagonal; ______

Siding WOOD; grade ______; type ______; size ______; exposure ______"; fastening galv. nails

Shingles ______; grade ______; type ______; size ______; exposure ______"; fastening galv. nails

Stucco ______; thickness ______"; Lath ______; weight ______ lb.

Masonry veneer ______ Sills ______ Lintels none Base flashing metal

Masonry: ☐ solid ☐ faced ☐ stuccoed; total wall thickness ______"; facing thickness ______"; facing material ______

Backup material ______; thickness ______"; bonding ______

Door sills ______ Window sills ______ Lintels ______ Base flashing ______

Interior surfaces: dampproofing, ______ coats of ______; furring ______

Additional information: ______

Exterior painting: material Jones Blair exterior paint; number of coats 3

Gable wall construction: ☒ same as main walls; ☐ other construction ______

6. FLOOR FRAMING:

Joists: wood, grade, and species W.C. const. fir; other ______; bridging 1x3; anchors 1/2"x12"

Concrete slab: ☒ basement floor; ☐ first floor; ☐ ground supported; ☐ self-supporting; mix 5 sack; thickness 3"

reinforcing 6/6-10/10 WWF; insulation ______; membrane polyurethane

Fill under slab: material gravel; thickness 4". Additional information ______

7. SUBFLOORING: (Describe underflooring for special floors under item 21.)

Material: grade and species 3/4" tung and groove; size 4x8; type CD

Laid: ☒ first floor; ☐ second floor; ☐ attic ______ sq. ft.; ☐ diagonal; ☒ right angles. Additional information: ______

8. FINISH FLOORING: (Wood only. Describe other finish flooring under item 21.)

Location	Rooms	Grade	Species	Thickness	Width	[illegible]	Finish
First floor							
Second floor							
Attic floor	sq. ft.						

Additional information ______

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DESCRIPTION OF MATERIALS

Figure 5-2 (continued)

DESCRIPTION OF MATERIALS

9. PARTITION FRAMING:

Studs: wood, grade, and species W C coast fir A size and spacing 2x4 @ 24" o.c. Other

Additional information Bearing Walls: 2x4 @ 16" o.c.

10. CEILING FRAMING:

Joists: wood, grade, and species Other Bridging

Additional information: truss (see attached detail)

11. ROOF FRAMING:

Rafters: wood, grade, and species Roof trusses (see detail): grade and species 45 0 pitch

Additional information: truss (see attached detail)

12. ROOFING:

Sheathing: wood, grade, and species ½" C.D. plywood ; ☒ solid, ☐ spaced " oc

Roofing asphalt , grade 235# ; size ; type

Underlay felt ; weight or thickness 15 , size ; fastening galv. nails

Built-up roofing ; number of plies , surfacing material

Flashing: material galv. metal ; gage or weight 30 , ☐ gravel stops, ☐ snow guards

Additional information.

13. GUTTERS AND DOWNSPOUTS:

Gutters: material galv. , gage or weight 26 ; size 12 ; shape moulded

Downspouts: material galv. ; gage or weight 26 ; size 3 ; shape square , number 4

Downspouts connected to: ☐ Storm sewer, ☐ sanitary sewer, ☐ dry-well ☒ Splash blocks material and size ~~redwood~~ Concrete

Additional information:

14. LATH AND PLASTER

Lath ☐ walls, ☐ ceilings material ; weight or thickness Plaster: coats ; finish

Dry-wall ☒ walls, ☒ ceilings material gyp. bd. ; thickness ½" ; finish texture

Joint treatment tape

15. DECORATING: *(Paint, wallpaper, etc.)*

Rooms	Wall Finish Material and Application	Ceiling Finish Material and Application
Kitchen	(1) prime &(2) enamel coats	same
Bath	" "	"
Other	(1) coat rubber base	"

Additional information applies only to finished areas (see plans)

16. INTERIOR DOORS AND TRIM:

Doors: type wood flush ; material Mahogany , thickness 1 3/8"

Door trim: type S. line ; material white pine Base: type S. line , material white pine , [illegible]

Finish: doors Fill, (2) coats stain ; trim (1) prime, (2) enamel coat

Other trim *(item, type and location)* window sills: formica

Additional information: closet doors metal bi-fold/louvered

17. WINDOWS: General Aluminum Corp. TARTAN TEL. 321-4316

Windows: type sliding , make Series 1800 , material aluminum ; sash thickness

Glass: grade insulated , ☐ sash weights, ☐ balances, type ; head flashing

Trim: type , material Paint , number coats

Weatherstripping: type , material Storm sash, number

Screens: ☐ full; ☒ half; type , number all , screen cloth material galv.

Basement windows: type sliding , material aluminum , screens, number all , Storm sash, number

Special windows

Additional information

18. ENTRANCES AND EXTERIOR DETAIL:

Main entrance door: material Mahogany ; width 36" ; thickness 1 3/4" Frame: material select fir , thickness 1 3/8

Other entrance doors: material Fir ; width 32" , thickness 1 3/8" Frame: material " , thickness "

Head flashing galv. metal Weatherstripping: type , saddles

Screen doors: thickness 1 ", number 1 , screen cloth material galv. Storm doors: thickness ", number

Combination storm and screen doors: thickness 1 "; number 1 , screen cloth material galv

Shutters: ☐ hinged, ☐ fixed. Railings , Attic louvers

Exterior millwork: grade and species Paint , number coats

Additional information

19. CABINETS AND INTERIOR DETAIL: Manufactured by Alpine cabinet Co.

Kitchen cabinets, wall units: material Timnath, Colo. , lineal feet of shelves see plans , shelf width 12"

Base units: material ; counter top formica , edging same

Back and end splash formica Finish of cabinets wood grain vinyl , number coats

Medicine cabinets: make ; model

Other cabinets and built-in furniture bath vanities per plans

Additional information

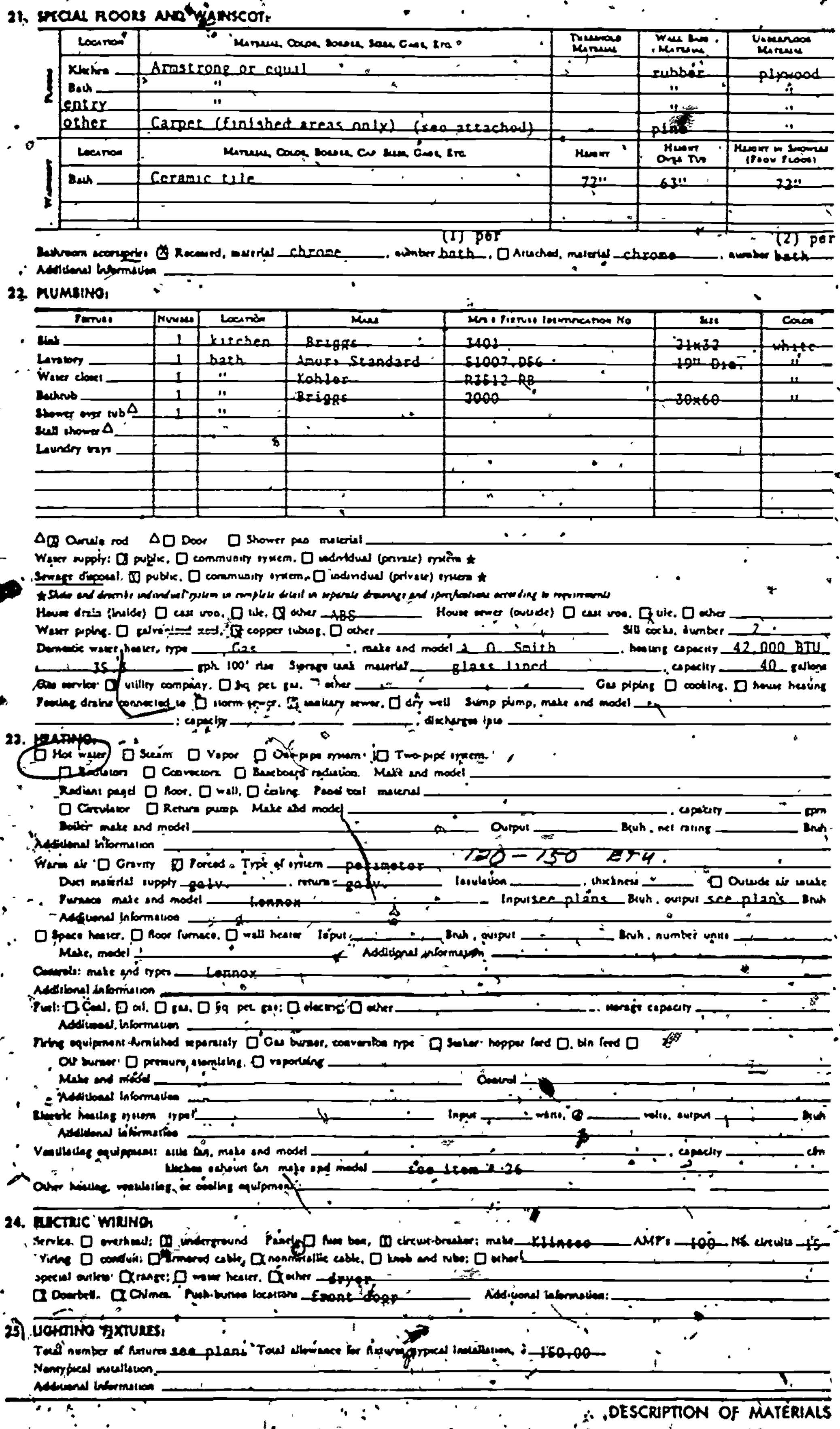
20. STAIRS:

Stair	Treads		Risers		Strings		Handrail		Balusters	
	Material	Thickness	Material	Thickness	Material	Size	Material	Size	Material	Size
Basement	fir	5/4	pine	3/4	W C fir	2x12	wood	2"	W.I.	3/4"
Main										
Attic										

Disappearing: make and model number

Additional information

Figure 5-2 (continued).



21. SPECIAL FLOORS AND WAINSCOT:

	Location	Material, Color, Border, Sizes, Gage, Etc.	Threshold Material	Wall Base Material	Underfloor Material
Floors	Kitchen	Armstrong or equil		rubber	plywood
	Bath	"		"	"
	entry	"		"	"
	other	Carpet (finished areas only) (see attached)	—	pine	"

	Location	Material, Color, Border, Cap Sizes, Gage, Etc.	Height	Height Over Tub	Height in Showers (From Floor)
Wainscot	Bath	Ceramic tile	72"	63"	72"

Bathroom accessories: ☒ Recessed, material chrome, number (1) per bath; ☐ Attached, material chrome, number (2) per bath

Additional information

22. PLUMBING:

Fixture	Number	Location	Make	Mfr's Fixture Identification No.	Size	Color
Sink	1	kitchen	Briggs	3401	21x32	white
Lavatory	1	bath	Amer. Standard	51007.056	19" Dia.	"
Water closet	1	"	Kohler	R3512 RB		"
Bathtub	1	"	Briggs	3000	30x60	"
Shower over tub△	1	"				
Stall shower△						
Laundry trays						

△☒ Curtain rod △☐ Door ☐ Shower pan material

Water supply: ☒ public, ☐ community system, ☐ individual (private) system ★

Sewage disposal: ☒ public, ☐ community system, ☐ individual (private) system ★

★*Show and describe individual system in complete detail in separate drawings and specifications according to requirements*

House drain (inside): ☐ cast iron, ☐ tile, ☒ other ABS House sewer (outside): ☐ cast iron, ☒ tile, ☐ other

Water piping: ☐ galvanized steel, ☒ copper tubing, ☐ other Sill cocks, number 2

Domestic water heater, type Gas, make and model A. O. Smith, heating capacity 42,000 BTU

35 gph. 100° rise. Storage tank material glass lined, capacity 40 gallons

Gas service: ☒ utility company, ☐ liq. pet. gas, ☐ other Gas piping: ☐ cooking, ☐ house heating

Footing drains connected to: ☐ storm sewer, ☒ sanitary sewer, ☐ dry well. Sump pump, make and model

; capacity ; discharges into

23. HEATING:

[✓ ☐ Hot water] ☐ Steam ☐ Vapor ☐ One-pipe system ☐ Two-pipe system.

☐ Radiators ☐ Convectors ☐ Baseboard radiation. Make and model

Radiant panel: ☐ floor, ☐ wall, ☐ ceiling. Panel coil: material

☐ Circulator ☐ Return pump. Make and model , capacity gpm

Boiler: make and model Output Btuh.; net rating Btuh.

Additional information

120 - 150 BTU.

Warm air: ☐ Gravity ☒ Forced. Type of system perimeter

Duct material: supply galv., return galv. Insulation , thickness ☐ Outside air intake

Furnace: make and model Lennox Input see plans Btuh.; output see plans Btuh.

Additional information

☐ Space heater; ☐ floor furnace; ☐ wall heater. Input Btuh.; output Btuh.; number units

Make, model Additional information

Controls: make and types Lennox

Additional information

Fuel: ☐ Coal; ☐ oil; ☐ gas; ☐ liq. pet. gas; ☐ electric; ☐ other ; storage capacity

Additional information

Firing equipment furnished separately: ☐ Gas burner, conversion type. ☐ Stoker: hopper feed ☐; bin feed ☐

Oil burner: ☐ pressure atomizing; ☐ vaporizing

Make and model Control

Additional information

Electric heating system: type Input watts; @ volts; output Btuh.

Additional information

Ventilating equipment: attic fan, make and model , capacity cfm.

kitchen exhaust fan, make and model see item # 26

Other heating, ventilating, or cooling equipment

24. ELECTRIC WIRING:

Service: ☐ overhead; ☒ underground. Panel: ☐ fuse box; ☒ circuit-breaker; make Klinsco AMP's 100 No. circuits 15

Wiring: ☐ conduit; ☐ armored cable; ☒ nonmetallic cable; ☐ knob and tube; ☐ other

Special outlets: ☒ range; ☐ water heater; ☒ other dryer

☒ Doorbell. ☒ Chimes. Push-button locations front door Additional information:

25. LIGHTING FIXTURES:

Total number of fixtures see plans. Total allowance for fixtures, typical installation, $ 150.00

Nontypical installation

Additional information

DESCRIPTION OF MATERIALS

Figure 5-2 (concluded)

DESCRIPTION OF MATERIALS

26. INSULATION:

Location	Thickness	Material, Type, and Method of Installation	Vapor Barrier
Roof			
Ceiling	6"	blown rock wool R-10 TIMBER LINE INSULATION	
Wall	3½"	batt R-11 482-1059	
Floor		482-3181	

HARDWARE: (make, material, and finish.) Stanley, brass, smooth
Privacy lock at master bedroom and bathrooms. keyed locks at all entrance doors including garage doors: all other doors passage knobs.

SPECIAL EQUIPMENT: (State material or make, model and quantity. Include only equipment and appliances which are acceptable by local law, custom and applicable FHA standards. Do not include items which, by established custom, are supplied by occupant and removed when he vacates premises or chattles prohibited by law from becoming realty.)

Garbage Disposal - Insinkerator Badger
Dishwasher - Frigidaire DW 3CDUU
Range - " RBE 353
Hood - Nautilus
Optional fireplace 'Heatilator Mark 123 Model 3036
Optional Medicine Cabinet Recessed Kent Model WAL 1420

27. MISCELLANEOUS: (Describe any main dwelling materials, equipment, or construction items not shown elsewhere; or use to provide additional information where the space provided was inadequate. Always reference by item number to correspond to numbering used on this form.)

Provide hot & cold water for washer
" 110 outlet for washer
" 220 outlet for dryer

PORCHES:

see plans

TERRACES:

see plans

GARAGES:

attached see plans

WALKS AND DRIVEWAYS:

Driveway: width 17', base material gravel, thickness 4"; surfacing material concrete, thickness 4".
Front walk: width 3', material concrete, thickness 4". Service walk: width ____ material ____, thickness ____".
Steps: material concrete; treads 12", risers 6". Cheek walls ____

OTHER ONSITE IMPROVEMENTS:

(Specify all exterior onsite improvements not described elsewhere, including items such as unusual grading, drainage structures, retaining walls, fence, railings, and accessory structures.)

LANDSCAPING, PLANTING, AND FINISH GRADING:

Topsoil 4" thick: ☒ front yard, ☒ side yards, ☒ rear yard to all feet behind main building.
Lawns (seeded, sodded, or sprigged): ☒ front yard sodded; ☐ side yards -; ☐ rear yard ____
Planting: ☐ as specified and shown on drawings; ☐ as follows:

____ Shade trees, deciduous, ____" caliper.		____ Evergreen trees, ____' to ____', B & B	
____ Low flowering trees, deciduous,	____' to ____'	____ Evergreen shrubs, ____' to ____', B & B	
____ High-growing shrubs, deciduous,	____' to ____'	____ Vines, 2-year ____	
____ Medium-growing shrubs, deciduous,	____' to ____'		
____ Low-growing shrubs, deciduous,	____' to ____'		

IDENTIFICATION.—This exhibit shall be identified by the signature of the builder, or sponsor, and/or the proposed mortgagor if the latter is known at the time of application.

Date 2[illegible] January 1975 Signature ____

Signature ____

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Figure 5-3

HEATING WORKSHEET
for Example Building

BUILDING SECTION	SIZE OR VOLUME	NET AREA OR VOLUME	U COEFF.	TEMP. DIFF. [68-(-9)]	HEAT LOSS	TOTALS
BEDROOM 1						
South wall	(15+3)x8	120	.07	77	647	
East wall	13.5x8	108	.07	77	282	
Windows (2)	3x4	24	.50	77	924	
Infiltration	2/3x15x13.6x8	1088	.018	77	1508	3361
BEDROOM 2						
East wall	14x8	112	.07	77	604	
North wall	11x8	76	.07	77	410	
Window	3x4	12	.50	77	462	
Infiltration	2/3x11x11x8	645	.018	77	894	2370
BATHROOM						
North wall	8x8	60	.07	77	323	
Window	2x2	4	.50	77	154	
Infiltration	3/4x7.5x11x8	495	.018	77	686	1163
BEDROOM 3						
North wall	12x8	84	.07	77	453	
West wall	10x8	80	.07	77	431	
Window	3x4	12	.50	77	462	
Infiltration	2/3x10x12x8	640	.018	77	887	2233
BEDROOM 4 & HALLWAY						
West wall	16x8	128	.07	77	690	
South wall	14x8	88	.07	77	474	
Window	2x3x4	24	.50	77	924	
Infiltration	2/3x14x16x8	1195	.018	77	1656	3744
LIVING ROOM						
South wall	32x8	203	.07	77	1094	
Door	3x7	21	.26	77	420	
Window	4x8	32	.62	77	1528	
East wall	13.5x8	108	.07	77	582	
Infiltration	2/3x19x13.5x8	1368	.018	77	1896	5520
DINING ROOM						
East wall	13.5x8	108	.07	77	582	
North wall	11x8	88	.07	77	474	
Infiltration	1/3x11x13.5x8	396	.018	77	549	1605

Figure 5-3 (continued)

HEATING WORKSHEET
for Example Building

BUILDING SECTION	SIZE OR VOLUME	NET AREA OR VOLUME	U COEFF.	TEMP. DIFF. [68-(-9)]	HEAT LOSS	TOTALS
KITCHEN, BREAKFAST						
North wall	18x8	122	.07	77	657	
Window	2.5x4	10	.50	77	385	
Window	3x4	12	.50	77	462	
Infiltration	1x18x11x8	1584	.018	77	2195	3699
FAMILY ROOM						
North wall	21.5x8	136	.07	77	733	
Patio door	6x6	36	.58	77	1608	
West wall	13x8	104	.20	77	1602	
South wall	22x8	176	.52	38	3478	
Infiltration	2x13x22x8	4576	.018	77	6342	13763
HALL						
West wall	17x8	136	.52	38	2687	
Infiltration	1x8x8x17	1088	.018	77	1508	4195
BASEMENT						
North wall	54x8	432	.10	23	994	
West wall	28x8	224	.10	23	515	
South wall	54x8	432	.10	23	994	
East wall	28x8	224	.10	23	515	
Floor	32x28	896	.10	23	2061	
Floor	13x22	286	.10	23	658	
Infiltration	1/6x54x13x8+ 1/6x15x32x8	1576	.018	77	2184	7921
CEILING						
Second floor	32x28	896	.04	77	2760	
Family room	13x22	286	.04	77	681	3641

TOTAL 53215

Figure 5-4

COOLING WORKSHEET
for Example Building

BUILDING SECTION	SIZE OR VOLUME	NET AREA OR VOLUME	U or UNIT HEAT GAIN	DDT	HEAT GAIN	TOTALS
BEDROOM 1						
South wall	18x8	120	.07	19	160	
East wall	13.5x8	108	.07	19	144	
Windows south	3x4	24	27		648	
Infiltration	1632	816	.018	14	205	1157
BEDROOM 2						
East wall	14x8	112	.07	19	149	
North wall	11x8	76	.07	19	101	
Window	3x4	12	27		324	
Infiltration	968	484	.018	14	122	696
BATHROOM						
North wall	8x8	60	.07	19	80	
Window	2x2	4	27		108	
Infiltration	660	660	.018	14	166	354
BEDROOM 3						
North wall	12x8	84	.07	19	112	
West wall	10x8	80	.07	19	106	
Window	3x4	12	27		324	
Infiltration	960	480	.018	14	121	663
BEDROOM 4 and HALLWAY						
West wall	16x8	128	.07	19	170	
South wall	14x8	88	.07	19	117	
Window	3x4	24	27		648	
Infiltration	1792	896	.018	14	226	1161
LIVING ROOM						
South wall	32x8	203	.07	19	270	
Door	3x7	21	.47	19	188	
Window	4x8	32	21		672	
East wall	13.5x8	108	.15	11	178	
Infiltration	2052	1026	.018	14	258	1566
DINING ROOM						
East wall	13.5x8	108	.07	19	144	
North wall	11x8	88	.07	19	117	
Infiltration	1188	198	.018	14	50	311

Figure 5-4 (continued)

COOLING WORKSHEET
for Example Building

BUILDING SECTION	SIZE OR VOLUME	NET AREA OR VOLUME	U or UNIT HEAT GAIN	DDT	HEAT GAIN	TOTALS
KITCHEN, BREAKFAST						
North wall	1848	122	.07	19	162	
Windows		22	27		594	
Infiltration	1584	1584	.918	14	399	1155
FAMILY ROOM						
North wall	21.5x8	136	.07	19	180	
West wall	13x8	104	.20	19	395	
South wall	22x8	176	.52	7	640	
Patio door	6x6	36	21		756	
Infiltration	2288	2288	.018	14	577	2548
HALL						
West wall	17x8	136	.52	7	495	
Infiltration	1088	1088	.018	14	274	769
CEILING						
Second floor	32x28	896	.04	39	1398	
Family room	13x22	286	.04	39	446	1844

TOTAL	12224
4 occupants x 225	900
Kitchen Appliances	1200
Total Sensible Heat Gain	14324
Latent Heat Gain (30%x14324)	4297
Latent + Sensible Heat Gain	18621
Cooling Load, Btu/hr	18621

No load is calculated for basement. No credit for cool basement taken.

Table 5-1. Surface Conductances and Resistances for Air Films; Conductance-Btu/(hr)(ft^2)(^{o}F) Resistance-(hr)(ft^2)(^{o}F)/Btu

ITEMS	WINTER		SUMMER	
	f_i	R_i	f_i	R_i
INTERIOR SURFACES				
Ceiling	1.63	0.61	1.08	0.92*
Sloped ceiling 45^o	1.60	0.62	1.32	0.76*
Walls and windows	1.46	0.68	1.46	0.68
Floor	1.08	0.92	1.08	0.92
EXTERIOR SURFACES				
Roofs, walls and windows	6.00	0.17+	4.00	0.25†

* Heat flow direction reversed from winter conditions

+ 15 mph wind

† 7.5 mph wind

Table 5-2. Resistance Values for Air Spaces (hr)(ft^2)(^{o}F)/Btu

ITEM \ Air Space	WINTER		SUMMER	
	3/4"	4"	3/4"	4"
Flat Roof	1.02	1.12	0.87	0.94
Wall	1.28	1.16	1.01	1.01

Table 5-3. Resistance Values for Building Materials (hr)(ft^2)(^oF)/Btu

TYPE AND MATERIAL		R
BUILDING BOARD		
Asbestos-cement:	1/8"	0.03
	1/4"	0.06
Gypsum:	3/8"	0.32
	1/2"	0.45
Plywood:	1/4"	0.31
	3/8"	0.47
	1/2"	0.62
	3/4"	0.93
Insulating Board	25/32"	2.06
Regular	1/2"	1.32
Laminated Paper	3/4"	1.50
Acoustic Tile	1/2"	1.25
	3/4"	1.89
Hardboard	3/4"	0.92
Particle Board	5/8"	0.82
Wood Subfloor	3/4"	0.94
MASONRY		
Concrete	6"	0.48
	8"	0.64
	10"	0.80
Concrete Blocks, 3 oval core		
Sand and Gravel	4"	0.71
	8"	1.11
	12"	1.28
Cinder	4"	1.11
	8"	1.72
	12"	1.89
Lightweight	4"	1.50
	8"	2.00
	12"	2.27
Concrete Blocks, 2 rect. core		
Sand and Gravel	8"	1.04
Lightweight	8"	2.18
Common Brick:	2"	0.40
	4"	0.80
Face Brick	2"	0.22
	4"	0.44
BUILDING PAPER		
15# felt		0.06

TYPE AND MATERIAL	R
SIDING	
Asbestos-cement	0.21
Wood shingles, 16"	0.87
Wood bevel, 1/2 x 8	0.81
Wood bevel 3/4 x 10	1.05
Wood plywood, 3/8	0.59
Aluminum or steel	0.61
Insulating Board:	
3/8" normal	1.82
3/8" foiled	2.96
FINISH FLOORING	
Carpet and fibrous pad	2.08
Carpet and rubber pad	1.23
Cork tile, 1/8"	0.28
Terrazzo, 1"	0.08
Tile, asphalt, linoleum, vinyl, rubber	0.05
Hardwood	0.08
INSULATION	
Blanket and Batt: 2-2 3/4"	7.00
3-3 1/2"	11.00
5 1/4-6 1/2"	19.00
Loose Fill	
Cellulose, per inch	3.70
Sawdust, per inch	2.22
Perlite, per inch	2.70
Mineral fibre (rock, slag, glass) 3"	9.00
4 1/2"	13.00
6 1/4"	19.00
7 1/2"	24.00
Vermiculite, per inch	2.20
ROOFING	
Asphalt	0.44
Wood	0.94
3/8" Built-up	0.33
Woods: oak, maple per inch	0.91
fir, pine, softwoods per inch	1.25
3/4"	0.94

Table 5-4. U Factors for Windows and Patio Doors
Btu/(hr)(ft^2)(^{o}F)

DESCRIPTION	WINTER
SINGLE GLASS	
Metal sash	1.13
Wood sash, 80% glass	0.02
DOUBLE GLASS:	
1/4" Air Space	
Metal sash	0.65
Wood sash, 80% glass	0.62
Wood sash, 60% glass	0.55
1/2" Air Space	
Metal sash	0.70
Wood sash, 80% glass	0.49
TRIPLE GLASS	
1/4" Air Space	
Metal sash	0.56
Wood sash, 80% glass	0.45
STORM WINDOWS	
1" to 4" Air Space	
Wood	0.50
Metal	0.56
SLIDING PATIO DOORS	
Single Glass	
Wood frame	1.07
Metal frame	1.13
Double Glass, 1/2" Air Space	
Wood frame	0.58
Metal	0.64

Table 5-5. U Factors for Solid Doors
Btu/(hr)(ft^2)(^{o}F)

THICKNESS (IN)	WINTER			SUMMER WITHOUT STORM DOOR
	WITHOUT STORM DOOR	WITH STORM DOOR, 50% GLASS		
		WOOD	METAL	
1	0.64	.030	0.39	0.61
1 ¼	0.55	0.28	0.34	0.53
1 ½	0.49	0.27	0.33	0.47
2	0.43	0.24	0.29	0.42

Table 5-6. Air Changes for Average Residential Conditions

KIND OF ROOM	AIR CHANGE PER HOUR	
	WINTER	SUMMER
Room with no windows or exterior doors	1/3	1/6
Rooms with windows or exterior doors on one side	2/3	1/2
Rooms with windows or exterior doors on two sides	1	2/3
Rooms with windows or exterior doors on three sides	1 1/3	1
Entrance halls and air locks	1 1/2	1

Table 5-7. Design Equivalent Temperature Differences (°F)

DESIGN OUTDOOR TEMPERATURE	85	95		105
TEMPERATURE RANGE DURING DAY	15-25	15-25	>25	>25
WALLS AND DOORS				
Wood frame and doors	14	24	19	29
Masonry	6	16	11	21
CEILINGS AND ROOF				
Under vented attic, dark roof	34	44	39	49
Built-up roof (no ceiling), light roof	26	36	31	41
FLOORS				
Over unconditioned rooms and open crawl space	5	15	10	20
Over basement, enclosed crawl space	0	0	0	0

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Table 5-8. Design Heat Gains Through Windows
Btu/(hr)(ft^2)

OUTDOOR DESIGN TEMPERATURE	SINGLE PANE			DOUBLE PANE		
	85	95	105	85	95	105
NO AWNINGS OR INSIDE SHADING						
North	23	31	38	19	24	28
Northeast; Northwest	56	64	71	46	51	55
East and West	81	89	96	68	73	77
Southeast; Southwest	70	78	85	59	64	68
South	40	48	55	33	38	42
WITH DRAPERIES OR VEN. BLINDS						
North	15	23	30	12	17	21
Northeast; Northwest	32	40	47	27	32	36
East and West	48	56	63	42	47	51
Southeast; Southwest	40	48	55	35	40	44
South	23	31	38	20	25	29
ROLLER SHADES, HALF DOWN						
North	18	26	33	15	20	24
Northeast; Northwest	40	48	55	38	43	47
East and West	61	69	76	54	59	63
Southeast; Southwest	52	60	67	46	51	55
South	29	37	44	26	32	36
AWNINGS						
North	20	28	35	13	18	22
Northeast; Northwest	21	29	36	14	19	23
East and West	22	30	37	14	19	23
Southeast; Southwest	21	29	36	14	19	23
South	21	28	35	13	18	22

Table 5-9. Shade Line Factors*
(5 hour average, 1 August)

WINDOW ORIENTATION	LATITUDE					
	25	30	35	40	45	50
East and West	0.8	0.8	0.8	0.8	0.8	0.8
Southeast; Southwest	1.9	1.6	1.4	1.3	1.1	1.0
South	10.1	5.4	3.6	2.6	2.0	1.7

* Multiply shade line factors by width of overhang to determine shadow line below overhang.

TRAINING COURSE IN
THE PRACTICAL ASPECTS OF
DESIGN OF SOLAR HEATING AND COOLING SYSTEMS
FOR
RESIDENTIAL BUILDINGS

MODULE 6

SIMPLIFIED DESIGN CALCULATIONS

SOLAR ENERGY APPLICATIONS LABORATORY
COLORADO STATE UNIVERSITY
FORT COLLINS, COLORADO

INTRODUCTION

TRAINEE-ORIENTED OBJECTIVE

To conduct a preliminary design study of a solar heating system.

SUB-OBJECTIVES

To put into practice the preliminary design methods presented during previous modules.

PROBLEM 1

We wish to estimate the size of collector array that should be used to provide approximately 75 percent of the space heating and service hot water loads for a house for which the heat load has been determined to be 17200 Btu/degree day. The house is to be built in your location. Use the Huck-Winn method.

PROBLEM 2

Now assume that, through conservation measures, the heat load is reduced to 15000 Btu/DD. Repeat the calculations.

PROBLEM 3

Repeat Problem 1 using the Balcomb-Hedstrom method.

TRAINING COURSE IN
THE PRACTICAL ASPECTS OF
DESIGN OF SOLAR HEATING AND COOLING SYSTEMS
FOR
RESIDENTIAL BUILDINGS

MODULE 7

DETAILED DESIGN METHODS

SOLAR ENERGY APPLICATIONS LABORATORY
COLORADO STATE UNIVERSITY
FORT COLLINS, COLORADO

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INTRODUCTION

TRAINEE-ORIENTED OBJECTIVE

The objective is to understand and utilize detailed performance estimation techniques for the determination of system performance.

SUB-OBJECTIVES

At the end of this module the trainee should be able to describe, explain, and utilize the Duffie-Beckman-Klein F-Chart procedure for the design of solar heating systems.

The simplified design techniques presented in Module 4 are not usually adequate for final design purposes because the collector performance is not adequately accounted for in those procedures. In this module, consideration is given to the design parameters of the collector itself in determining system performance, and is devoted primarily to the selection of the collector array. The methods presented in Module 16 for the selection of other components, such as pumps, heat exchangers, etc., are valid for final design purposes.

THE DUFFIE-BECKMAN-KLEIN PROCEDURE

This material is based upon the work presented in Reference 1. In that paper, a general design procedure for solar heating systems was developed based upon information obtained from many simulations of solar heating systems utilizing a detailed simulation program. The result was a simple graphical method using monthly average meteorological data which may be used for the design of solar heating systems.

THE APPROACH

The approach taken in Reference 1 was to use a simulation program to develop a generalized performance chart. This generalized performance chart correlates long-term performance of solar heating systems with system design parameters, building construction, and weather. This chart may then be combined with cost figures to provide a method by which architects and heating engineers can determine the economic optimal design of residential space and water heating systems.

The water system considered is shown in Figure 7-1. This analysis is applicable to systems using liquid as the transport medium and for storage. The mathematical models for the system components that were used in the simulation studies are presented below.

COLLECTOR MODEL

The method of Hottel and Whillier (Ref. 5) was used to model the collector. The equation for the useful energy collected is

$$Q_u = F_R A \left[H_T \, \tau\alpha - U_L (T_i - T_a)\right] = (\dot{m}\, Cp)_c (T_o - T_i) \qquad (7\text{-}1)$$

where

$$F_R = \frac{(\dot{m}\, Cp)_c}{A\, U_L} \left[1 - \exp\left(\frac{F' U_L A}{(\dot{m}\, Cp)_c}\right)\right] \qquad (7\text{-}2)$$

The terms in this equation are defined below:

F' = collector efficiency factor

A = collector area

H_T = radiation on the collector

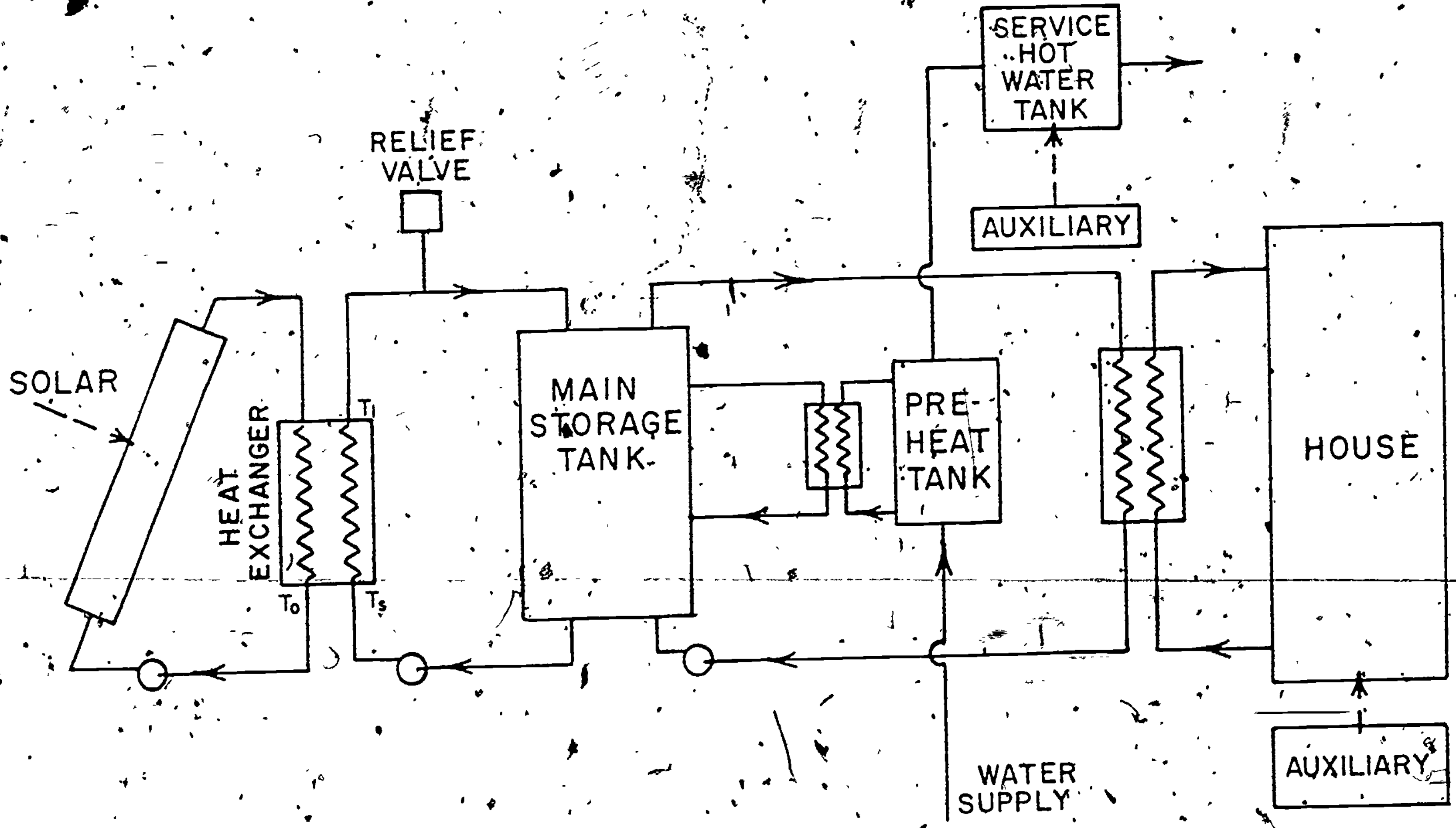
SERVICE HOT WATER TANK
RELIEF VALVE
AUXILIARY
SOLAR
HEAT EXCHANGER
T_I
T_o
T_s
MAIN STORAGE TANK
PRE-HEAT TANK
HOUSE
WATER SUPPLY
AUXILIARY

Figure 7-1. Schematic Diagram of a Liquid-Based Solar Space and Water Heating System

$\overline{\tau\alpha}$ = product of cover transmittance and collector plate absorptance

U_L = collector loss coefficient

T_i = collector inlet temperature

T_a = ambient temperature

$(\dot{m}\,Cp)_c$ = collector fluid capacitance rate

T_o = collector outlet temperature

STORAGE MODEL

The heat balance equation for the storage system is

$$C_S \frac{dTs}{dt} = Q_D - (Q_L-E_L) - (Q_W-E_W) \tag{7-3}$$

where

C_s = heat capacity of storage

T_s = storage temperature

Q_L = space heating load

E_L = auxiliary energy required to meet the space heating load

Q_W = service hot water heating load

E_W = auxiliary energy required to meet the service hot water load

Q_D = energy transferred to storage tank

HEAT EXCHANGER MODEL

The thermal performance of the heat exchanger was modeled according to the equation

$$Q_{hx} = \varepsilon_c (\dot{m}\,Cp)_{min} (T_0 - T_s) = (\dot{m}\,Cp)_s (T_1 - T_s) \qquad (7\text{-}4)$$

where T_s = temperature of water in storage

$(\dot{m}\,Cp)_{min} = \min\,[(\dot{m}\,Cp)_c, (\dot{m}\,Cp)_s]$

T_0 and T_1 are the temperatures at the outlets of the heat exchanger as indicated on Figure 7-1. Thermal losses in the piping were disregarded in the analysis. This is valid if pipes are well insulated and not excessively long.

The equations for the heat exchanger and collector were combined to give

$$Q_u = F_R' A [H_T \overline{\tau\alpha} - U_L (T_s - T_a)] \qquad (7\text{-}5)$$

where

$$F_R' = F_R \left\{ 1 + \left[\frac{F_R U_L A}{(\dot{m}\,Cp)_c}\right] \left[\frac{(\dot{m}\,Cp)_c}{\varepsilon_c (\dot{m}\,Cp)_{min}} - 1\right]\right\}^{-1} \qquad (7\text{-}6)$$

The term, F_R'/F_R is considered to be a "heat exchanger factor". Its value lies between 0 and 1 and represents a penalty paid for the use of the heat exchanger.

The term in the storage model, Q_D, was then expressed in terms of the above relations according to

$$Q_D = \begin{cases} F_R' A [H_T \tau\alpha - U_L (T_s - T_a)], & T_1 \leq T_{max} \\ (\dot{m}\,Cp)_s (T_{max} - T_s), & T_1 > T_{max} \end{cases} \qquad (7\text{-}7)$$

where T_{max} is 212 °F

LOAD MODEL

The house heating load was modeled as

$$Q_L = UA\ (T_R - T_a) \qquad (7\text{-}8)$$

where T_R is the room temperature and UA is the space heating load at design conditions divided by the design temperature difference.

The service water heating load was modeled by

$$Q_w = (\dot{m}\ Cp)_w\ (T_w - T_m) \qquad (7\text{-}9)$$

where T_w represented the minimum acceptable hot water temperature and T_m represented the temperature from the cold water supply lines.

AUXILIARY ENERGY

The auxiliary energy was modeled in two parts. The first represented the amount of energy required for service hot water, E_w, while the second represented the amount required for space heating, E_L. These were modeled according to

$$E_w = (\dot{m}\ Cp)_w\ (T_w - T_s) \qquad (7\text{-}10)$$

and

$$E_L = [Q_L - Q_{max} - (UA)_s\ (T_s - T_a)] \qquad (7\text{-}11)$$

where

$$Q_{max} = \varepsilon_L C_{min}\ (T_s - T_R)/UA. \qquad (7\text{-}12)$$

The term C_{min} represents the minimum of the two fluid flow capacitances through the load heat exchanger and is usually that of air. The term $\varepsilon_L C_{min}/UA$ provides a measure for sizing the load heat exchanger.

SENSITIVITY ANALYSIS

The development of the general design procedure for solar heating systems began with an extensive study of the effects of various design parameters on the long-term system performance. The sensitivity study indicated three parameters upon which the performance was highly sensitive whereas additional system parameters had negligible effect on performance relative to these three highly sensitive parameters. These parameters are summarized as follows:

COLLECTOR FLUID CAPACITANCE RATE

The product of the mass flow rate and the specific heat of the transport medium through the collector strongly affects the heating system performance. Duffie and Beckman (Ref. 2) have shown that the optimal collector fluid capacitance rate is infinitely large. However, the dependence of system performance on the collector capacity rate is asymptotic and only a small gain in energy collection rate is realized if the collector fluid capacitance rate per unit area is increased when the collector flow factor (defined below) exceeds 10.

$$F' = F_R'/F_R = \frac{(\dot{m}\ Cp)_c}{U_L\ A\ F'} \left[1 - \exp\left(\frac{-U_L F' A}{(\dot{m}\ Cp)_c} \right) \right]. \qquad (7\text{-}13)$$

Low collector fluid capacitance rates, that is, rates for which F' is less than 5, also result in reductions of energy collection; the nominal value for collector fluid capacitance rate was selected as $210\ \mathrm{kJ\ hr^{-1}\ {}^{o}C^{-1}\ m^{-2}}$ ($10.28\ \mathrm{Btu\text{-}hr^{-1}\ {}^{o}F^{-1}\ Ft^{-2}}$) for the study.

STORAGE CAPACITY

In an economic study of solar heating systems, Löf and Tybout (Ref. 3) concluded that the storage tank capacity that resulted in minimum cost solar heating was in the range of 200 to 300 kJ/°C (50-75 kg of stored water) per square meter of collector area. That is the equivalent of 1.25 to 1.84 gallons of water per square foot of collector area. They also indicated that the performance of solar heating systems is rather insensitive to the amount of storage capa[illegible]ithin this general range. These results are valid so long as one is not considering seasonal storage, that is, storing summer heat for use in the winter. Results from simulations for several different storage capacities are in general agreement with the Löf and Tybout study and are shown in Figure 7-2. Consequently, a storage capacity of 80 kg water/sq. meter (2 gallons water/sq. ft.) was used in the study to develop the generalized performance curves.

LOAD HEAT EXCHANGER SIZE

A dimensionless parameter ($\varepsilon_L C_{min}/UA$), where ε_L is the effectiveness of the space heating load heat exchanger, C_{min} is the minimum capacitance rate in the load heat exchanger, and UA represents a constant characterizing space heating load, was found to provide a measure of the size of heat exchanger needed in order to supply heat to a specified building. Figure 7-3 indicates how the performance of a space heating system is related to this parameter. For values of this parameter less than 1, the reduction in system performance due to having a heat exchanger that is too small will be appreciable. Values for this parameter between 1 and 3 should be used for optimal performance. Consequently, the generalized performance charts were developed utilizing a value of this parameter equal to 2.

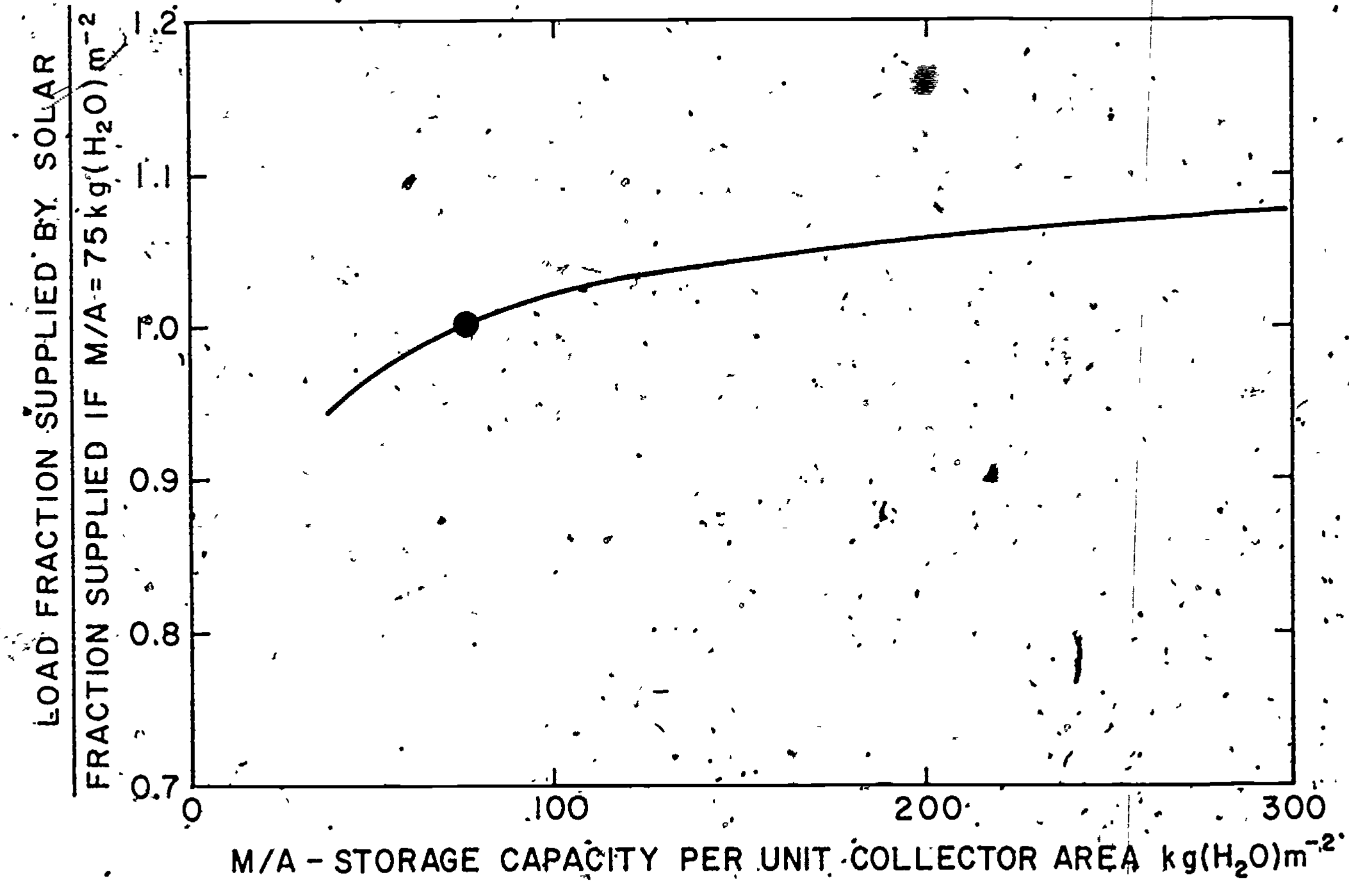

Figure 7-2. (from Reference 1)

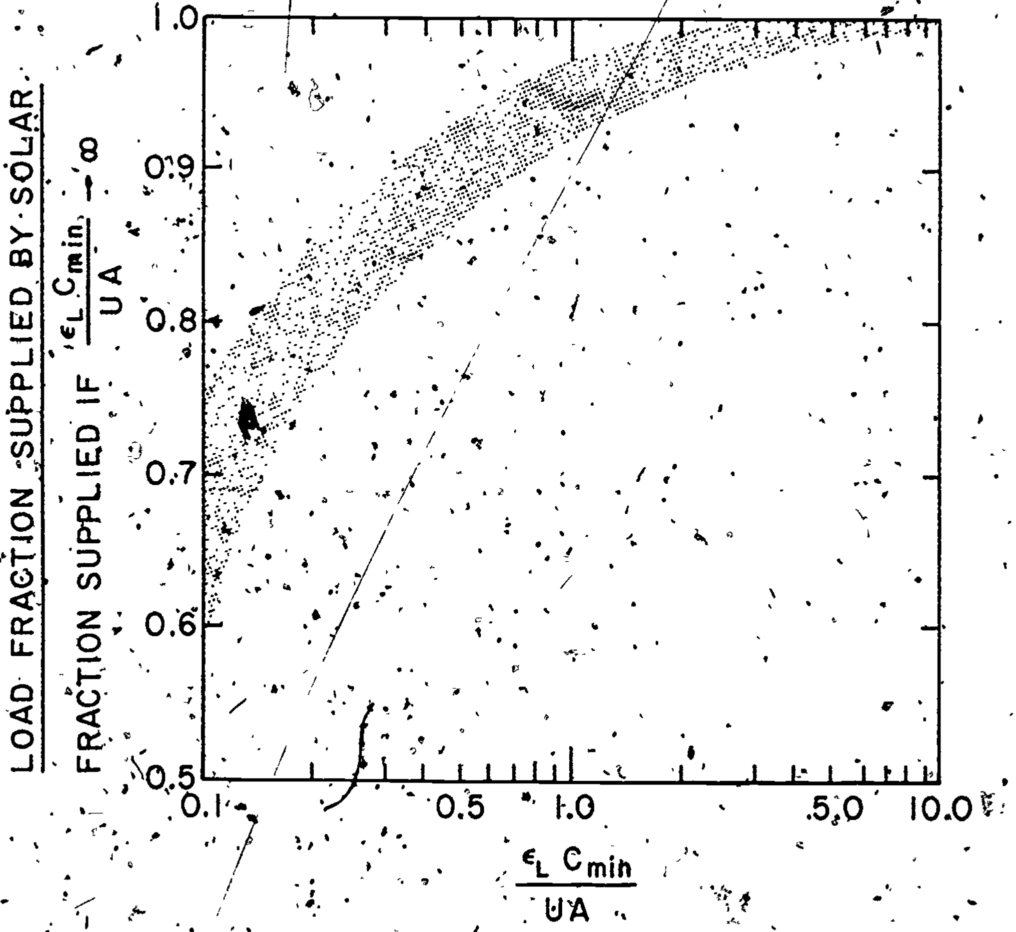

Figure 7-3. (from Reference 1)

CORRELATION OF SYSTEM PERFORMANCE

The correlation of f, the fraction of the monthly load supplied by solar, to various system parameters was investigated by making more than 300 simulation studies, each of which estimated month by month system performance by performing calculations at half-hour intervals using meteorological data for an average year for Madison, Wisconsin. The results of these studies are shown graphically in Figure 7-4. This chart may be used to estimate the performance of a solar heating system, month by month, as a function of the system design and local weather conditions.

Although the f curves shown in Figure 7-4 were developed using simulation results for Madison, Wisconsin, climatological data studies have indicated that the performance chart can be applied for other locations with satisfactory results.

APPLICATION OF THE PERFORMANCE CHART TO A SYSTEM DESIGN

The application of the performance chart on Figure 7-4 will be illustrated in this section by considering the design of a residential-type structure in Indianapolis, Indiana. Suppose that the design heat load is found to be 61,000 Btu/hr at an ambient temperature of 4 °F. This results in a 24,000 Btu/degree-day house. The monthly and annual heating loads may be determined by referring to Table 4-5 to obtain an estimate of the degree-days/month as well as the annual degree days for Indianapolis, Indiana. Suppose that the service water heating load is estimated to be 80 gallons per day to be raised from 52 °F to 140 °F, and this does not change throughout the year.

Flat-plate collectors are to be used, and the design characteristics of these collectors, as determined from a collector performance curve, are

$$F_R U_L = 0.86 \text{ BTU/hr-Ft}^2 - {}^oF$$

$$F_R \overline{\tau\alpha} = 0.72$$

Also suppose that the other required system parameters are

$$(\dot{m}\ Cp)_C/A = 10.28 \text{ BTU/hr-Ft}^2 - {}^oF$$
$$(\dot{m}\ Cp)_S/A = 12.05 \text{ BTU/hr-Ft}^2 - {}^oF$$
$$\varepsilon_C = 0.70.$$

Then

$$F_R'/F_R = \left\{ 1 + \left(\frac{0.86}{10.28}\right)\left(\frac{1}{.7} - 1\right)\right\}^{-1} = 0.965$$

Therefore

$$F_R' U_L = 0.83$$

and

$$F_R' \overline{\tau\alpha} = 0.69$$

The collectors are to be mounted facing due south at a slope equal to the latitude. The ratio of the storage tank to the collector area is to be 80 kg water/sq. meter of collector (1.64 lb/Ft^2).

The calculations are shown in Tables 7-1(a), and 7-1(b). The values for $\overline{H}$ and $\overline{K_T}$ were obtained from Table 3-1. The values for $\overline{R}$ were obtained from Figure 3-26. $\overline{H_T}$ is obtained from $\overline{R} \cdot \overline{H}$. The degree

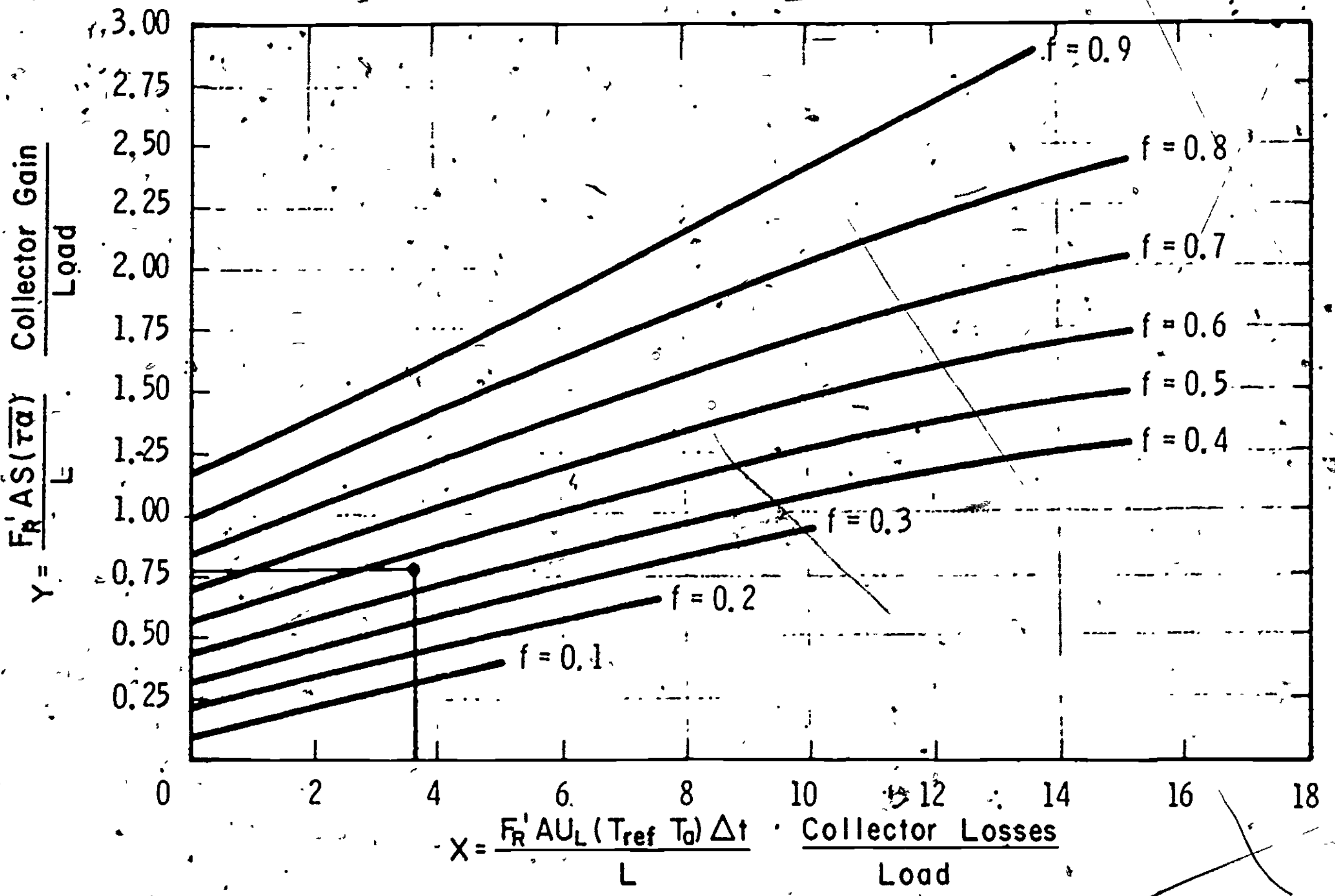

Figure 7-4. f-Chart for Liquid-Based Solar Heating Systems

NOMENCLATURE

A	collector area [m^2]
f	fraction of the total heating load supplied by solar energy each month
F_R	collector efficiency factor
F_R'	a modified collector efficiency factor which accounts for the penalty in energy collection imposed by the use of a double-loop flow circuit $$F_R' = F_R \left[\frac{1}{1 + \frac{F_R U_L A}{(\dot{m}C_p)_c}\left[\frac{\text{[illegible]}_c}{\text{[illegible]}_{min}\epsilon_c} - 1\right]} \right]$$
L	total space and water heating loads for each calendar month [kJ]
$(\dot{m}C_p)_c$	collector fluid capacitance rate [kJ hr^{-1} C^{-1}]
$(\dot{m}C_p)_{min}$	minimum fluid capacitance rate through the collector tank heat exchanger [kJ hr^{-1} C^{-1}]
S	total radiation per unit area incident upon the collector during each calendar month [kJ m^{-2}]
T_{ref}	reference temperature [100 C]
U_L	collector overall energy loss coefficient [kJ hr^{-1} C^{-1} m^{-2}]
Δt	number of hours in each calendar month [hr]
ϵ_c	effectiveness of the collector-storage tank heat exchanger
$(\overline{\tau\alpha})$	average product of the cover transmittance and the collector plate absorptance

TABLE 7-1(a). - Load and Meteorological Data

MONTH	$\overline{H}$ Btu/FT²	$\overline{K}_T$	$\overline{R}$	$\overline{H}_T$ Btu/Ft²	OD	$\overline{T}_A$ °F	$(T_{REF}-\overline{T}_a)$	L_H Btu 10^{-6}	L Btu 10^{-6}	S Btu/Ft²
JAN	526	.380	1.58	831	1113	29	183	26.71	28.45	25761
FEB	797	.424	1.47	1172	949	31	181	22.78	24.52	32816
MAR	1184	.472	1.23	1456	809	39	173	19.42	21.16	45136
APR	1481	.470	1.04	1540	432	51	161	10.37	12.11	46200
MAY	1828	.511	0.93	1700	177	59	153	4.25	5.99	52700
JUN	2042	.543	0.89	1797	39	64	148	0.94	2.68	53910
JUL	2040	.554	0.90	1836	0	65	147	0	1.74	56916
AUG	1836	.552	0.99	1814	0	65	147	0	1.74	56234
SEP	1513	.549	1.16	1755	90	62	150	2.16	3.90	52650
OCT	1094	.520	1.42	1553	316	55	157	7.58	9.32	48143
NOV	662	.413	1.54	1019	723	41	171	17.35	19.09	30570
DEC	491	.391	1.85	908	1051	31	181	25.22	26.96	28148
									157.66	

TABLE 7-1(b). Fraction of Load Supplied by Solar

MONTH	$F_R'U_L(T_{REF}-\overline{T}_a)$ $\Delta t/L$ FT^{-2}$\cdot 10^{-3}$	$F_R'\overline{\tau\alpha}$ S/L FT^{-2}$\cdot 10^{-4}$	X and Y Values						f-Fraction of Load Supplied by Solar		
			500Ft²		750Ft²		1000Ft²		500	750	1000
			X	Y	X	Y	X	Y			
JAN	3.97	6.25	1.985	.3125	2.98	.47	3.97	.63	.18	.30	.35
FEB	4.12	9.23	2.06	.46	3.09	.69	4.12	.92	.35	.47	.58
MAR	5.05	14.72	2.53	.74	3.79	1.10	5.05	1.47	.54	.67	.85
APR	7.94	26.32	3.97	1.32	5.96	1.97	7.94	2.63	.78	.95	1.00
MAY	15.77	60.71	7.89	3.04	11.83	4.55	15.77	6.07	1.00	1.00	1.00
JUN	33.00	138.80	16.50	6.94	24.75	10.41	33.00	13.88	1.00	1.00	1.00
JUL	52.17	225.70	26.08	11.29	39.12	16.92	52.17	22.57	1.00	1.00	1.00
AUG	52.17	223.00	26.08	11.15	39.12	16.72	22.98	22.30	1.00	1.00	1.00
SEP	22.99	93.15	11.49	4.66	17.24	6.99	10.40	9.32	1.00	1.00	1.00
OCT	10.40	35.64	5.20	1.78	7.80	2.67	5.35	3.56	.92	1.00	1.00
NOV	5.35	11.05	2.68	.55	4.01	.83	4.15	1.11	.33	.50	.60
DEC	4.15	7.20	2.08	.36	3.11	.54	[illegible]	.72	.20	.32	.39
								YEARLY FRACTION	.45	.57	.64

$$\text{YEARLY FRACTION} = \frac{\Sigma \text{ MONTHLY LOADS BY SOLAR}}{\Sigma \text{ MONTHLY LOADS}}$$

days were determined from Table 4-5. The average ambient temperatures were determined from knowing the degree days per month and applying the equation

$$\overline{T}_a = 65 - DD/n \qquad (7\text{-}14)$$

where n is equal to the number of days in the month. The heating load, L_H, is obtained from the product of the degree days and the design heating load for the house.

A service hot water load was included in the analysis. It was assumed that the service hot water load would require 80 gallons per day to be raised from 52 °F to 140 °F. It was assumed that this load was constant throughout the year. Thus,

$$L_{SHW} = (80\ \frac{gal}{day})\ (8.25\ \frac{lb}{gal})\ (\frac{1\ Btu}{lb\text{-}^oF})\ (140\text{-}52)^oF\ (30\ days)$$

$$= 1{,}742{,}400\ BTU/month.$$

This value was added to the heating load to obtain the total load, L, for each month.

The value for the total radiation on each square foot of the collector for each month, S, is obtained from the product of H_T and the number of days in each month.

The abscissa and ordinate (X and Y) values for the f-chart, Figure 7-4, were calculated for collectors of 500, 750, and 1000 Ft^2 using the values calculated from above. The fraction of the load, f, supplied by solar for each month was determined from Figure 7-4 using the calculated values for X and Y. The figures shown on the bottom row of the f-column on Table 7-1(b) represent the fraction of the annual load supplied by solar for each size of collector array considered.

The f-chart curves were developed using nominal values for storage capacity, collector-to-storage heat exchanger performance, and load heat exchanger performance. The corrections that should be made to X and Y for systems that vary from the nominal values used in the development of the curves are given in Figures 7-5 through 7-7: Figure 7-5 may be used to determine the correction to X for variations in the storage size from the nominal value of 2 gallons of water per square foot of collector. Figure 7-6 may be used to determine the correction to Y for variations in the load heat exchanger factor from the nominal value of 2. Figure 7-7 may be used to determine the value of F_R'/F_R in terms of the collector heat exchanger factor.

A similar study was conducted for an air heating system shown in Figure 7-8. The resulting f-chart is shown in Figure 7-9. The axes are similar to Figure 7-4, except there is no heat exchanger from collectors to storage and ε_c is taken to be 1.0. The correction factors are shown in Figures 7-5 and 7-10. Figure 7-5 may be used to determine the correction to X that should be applied to systems having storage sizes that differ from the nominal value of 0.75 cubic feet of rocks per square foot of collector. Figure 7-10 may be used to determine the correction to X that should be applied to systems having collector air flow rates that vary from the nominal value of two standard cubic feet per minute (SCFM) per square foot of collector.

For either air or water system, the values for $F_R \overline{\tau\alpha}$ and $F_R U_L$ may be obtained in the manner illustrated by Figure 7-11 if test data for a given collector are available. All that is necessary is to plot a straight line through the data and read $F_R \overline{\tau\alpha}$ from the y-intercept. The value for $F_R U_L$ is determined by the slope of the straight line.

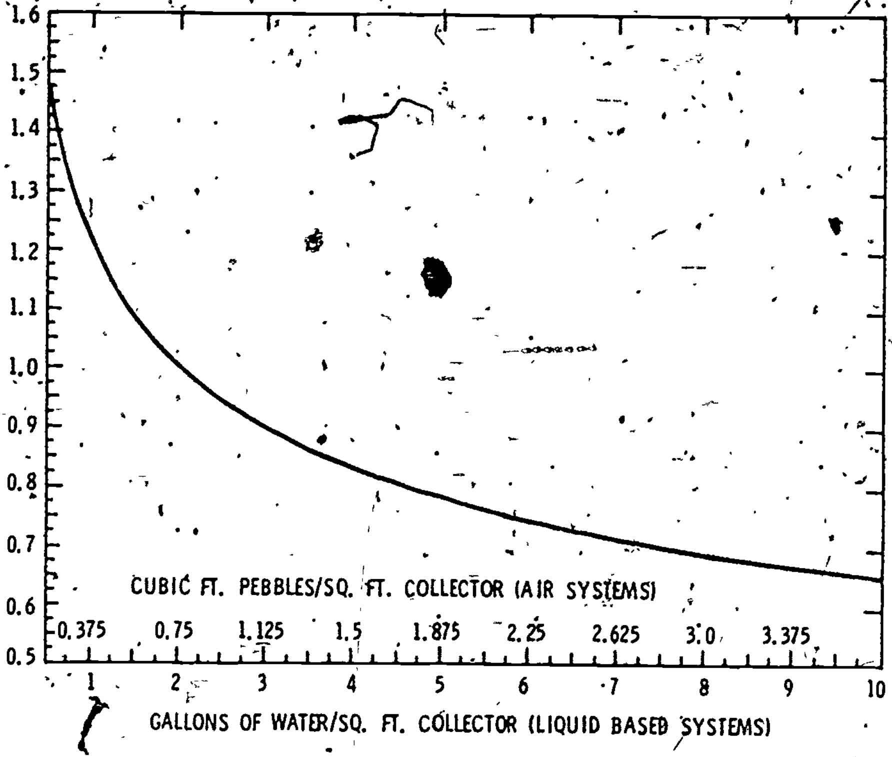

Figure 7-5. Storage Size Correction Factor

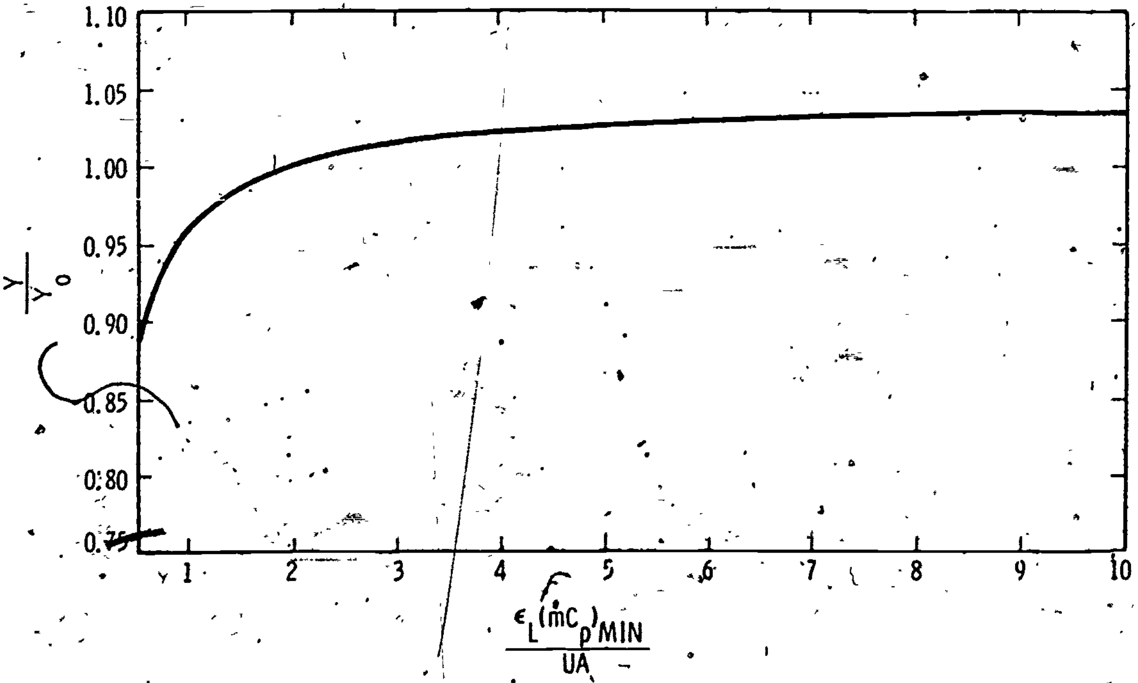

Figure 7-6. Load Heat Exchanger Correction Factor

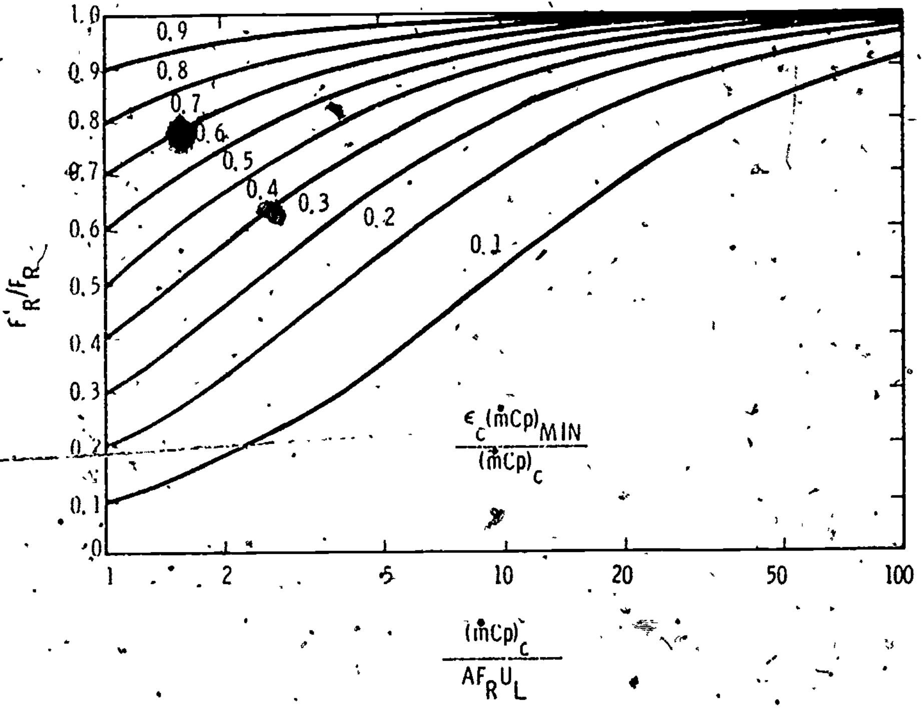

Figure 7-7. Collector Heat Exchanger Correction Factor

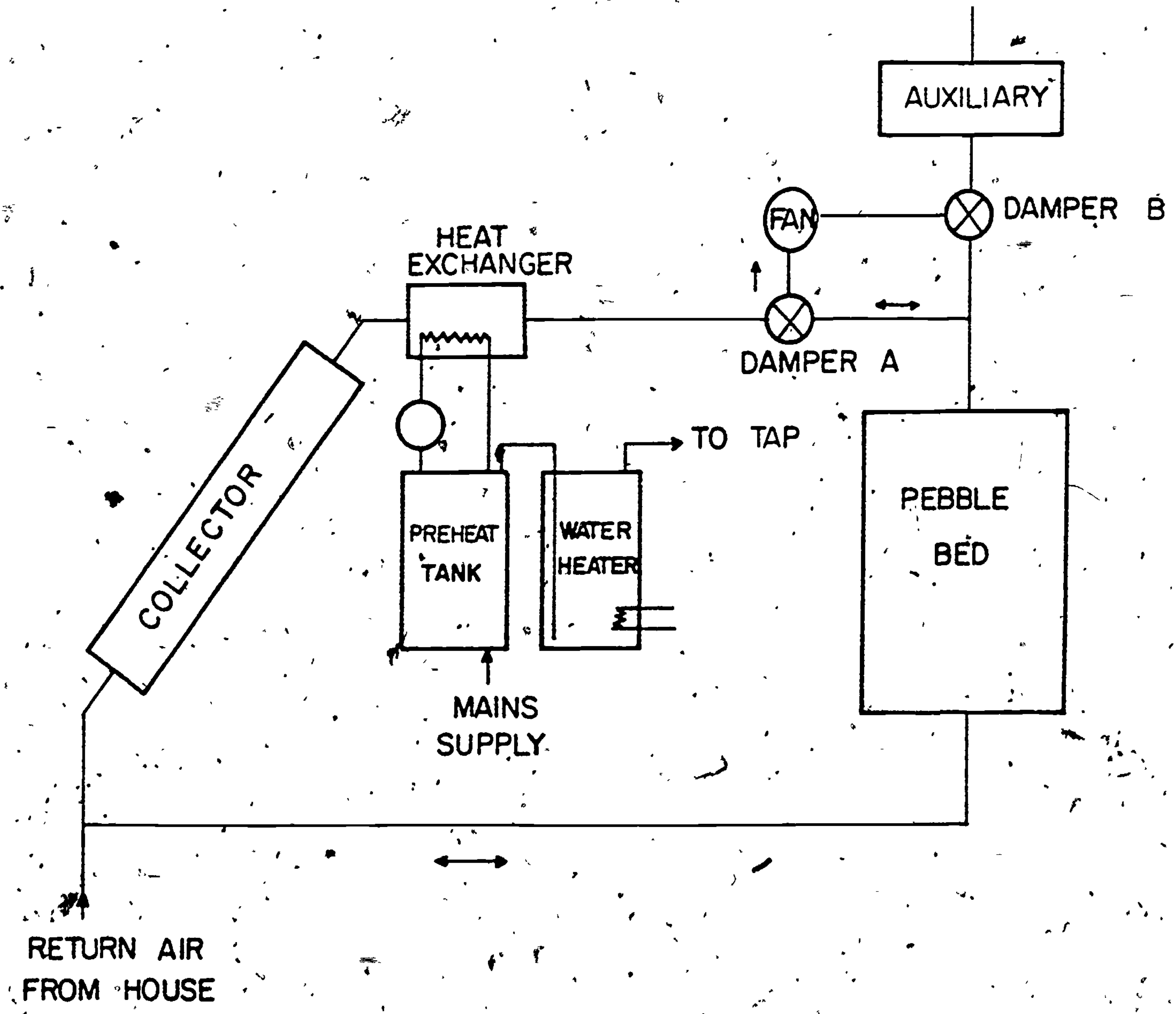
AUXILIARY
FAN
DAMPER B
HEAT EXCHANGER
DAMPER A
COLLECTOR
PREHEAT TANK
WATER HEATER
TO TAP
PEBBLE BED
MAINS SUPPLY
RETURN AIR FROM HOUSE

Figure 7-8. Schematic Diagram of an Air-Based Solar Space and Water Heating System

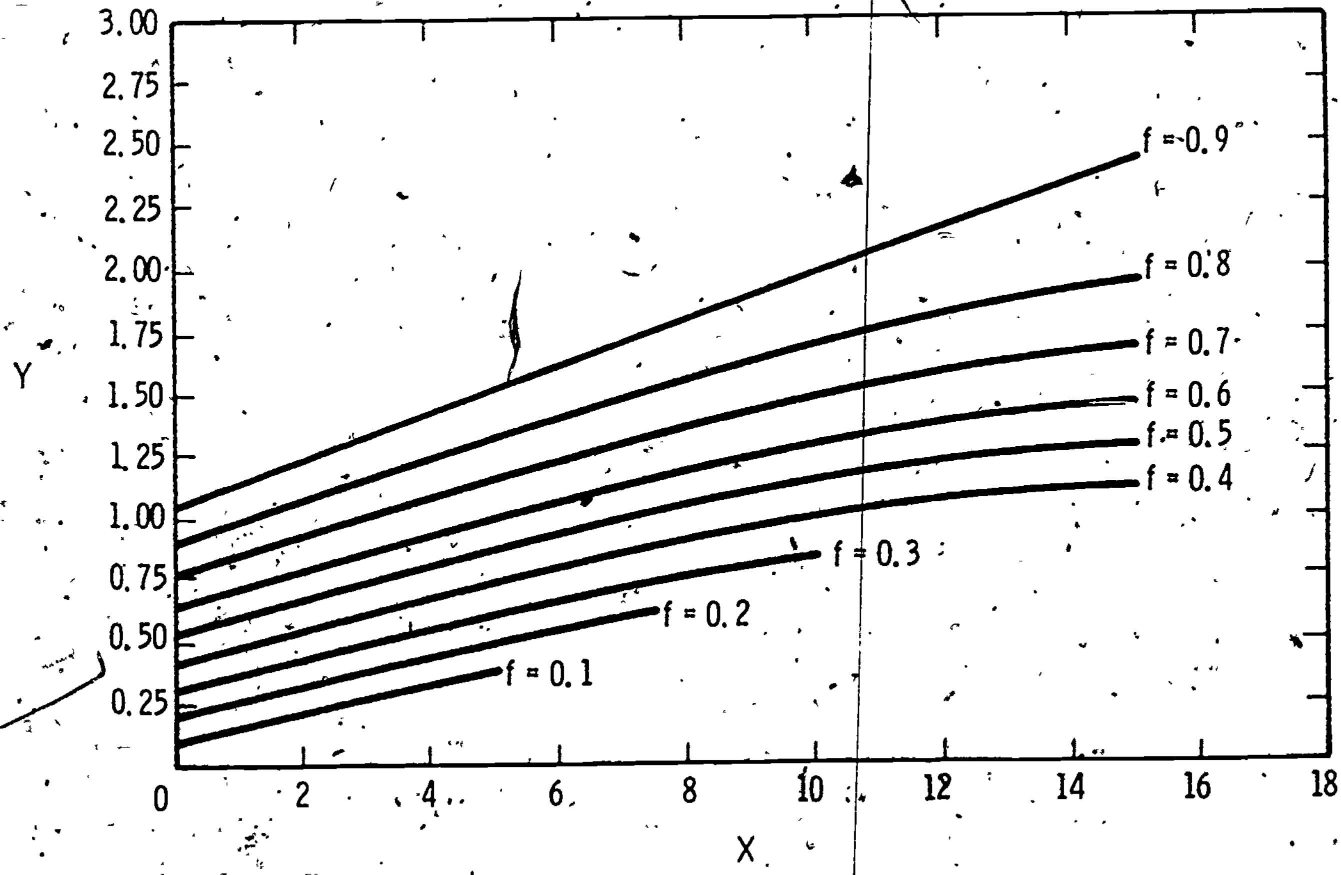

Figure 7-9. f-Chart for Solar Air Heating Systems

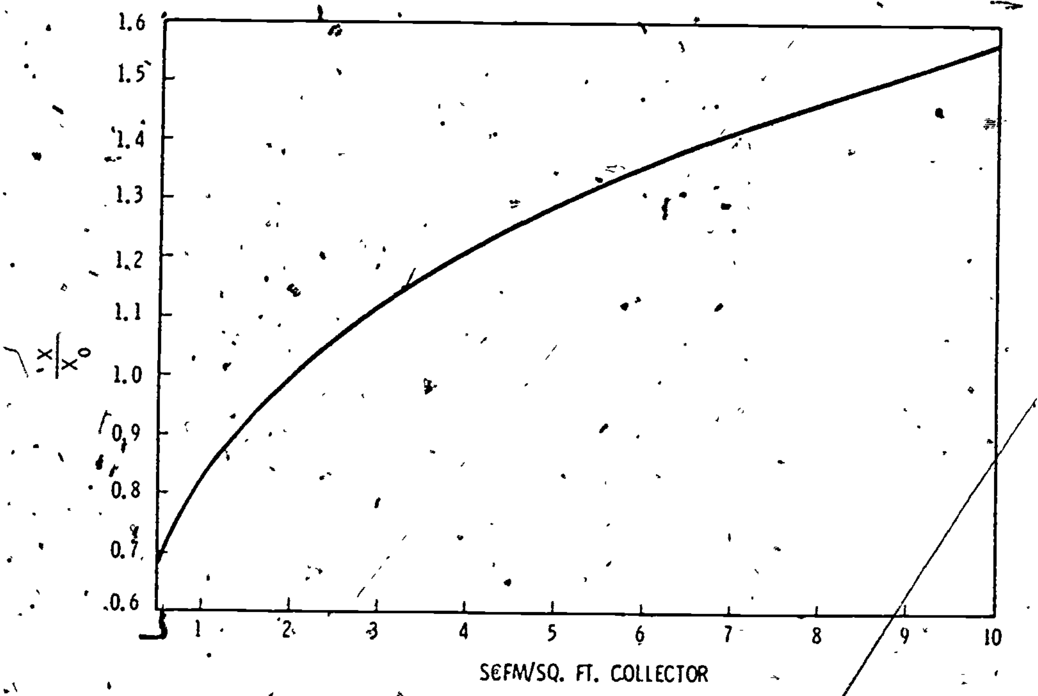

Figure 7-10. Collector Air Flow Rate Correction Factor

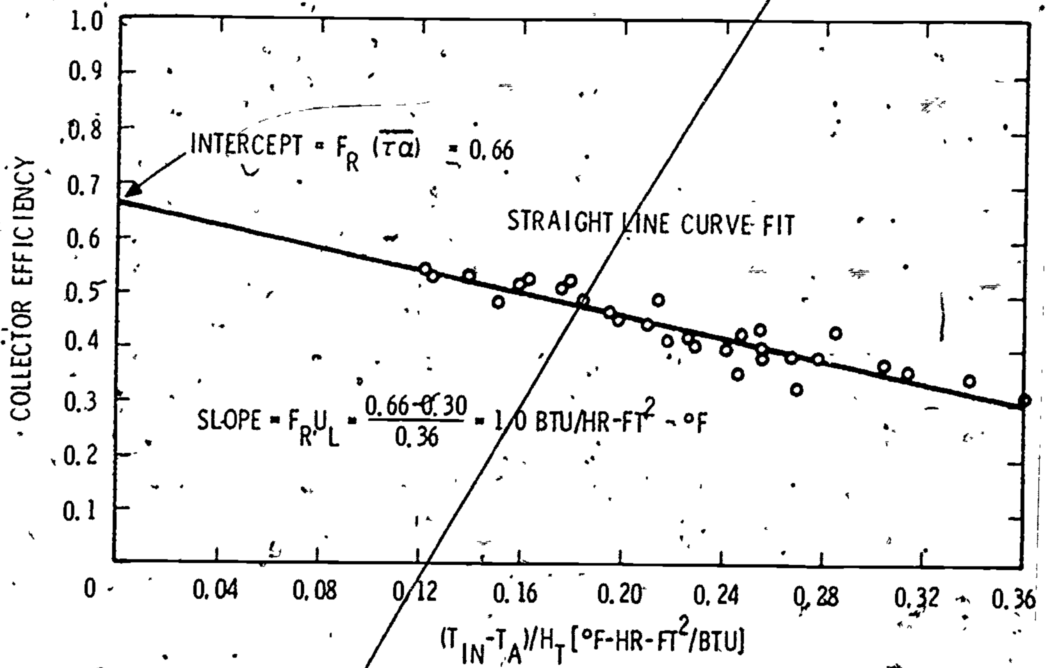

Figure 7-11. Experimental Collector Performance

AUTOMATED USE OF THE PERFORMANCE CHARTS

The calculations required for use of the performance charts are tedious; the efficiency of the calculation process can be improved significantly by automating the calculations. By means of standard curve-fitting programs the following relationships have been obtained for the f-charts for water and air systems, respectively (Ref. 4).

$$f_w = 1.029Y - 0.065X - 0.245Y^2 + 0.0018X^2 + 0.0215Y^3 \qquad (7\text{-}15)$$

$$f_a = 1.04Y - 0.065X - 0.159Y^2 + 0.00187X^2 - 0.0095Y^3 \qquad (7\text{-}16)$$

These equations may be used instead of the f-charts to obtain appropriate values for f.

A similar equation has been developed for potable hot water systems having the configurations shown in Figures 7-12(a) and 7-12(b). The air or water equations may be used, but a correction to X must be made. The correction is given as

Service Hot Water Only;

$$X = \frac{AF_R'U_L\Delta t\left[11.6 + 1.18\,T_w + 3.86\,T_m - 2.32\,\bar{T}_A\right]}{L_{SHW}}$$

T_w is hot outlet temperature typically 140 °F

T_m is cold inlet temperature typically 50 °F

$\bar{T}_A$ is the average outdoor air temperature

L_{SHW} is service hot water load

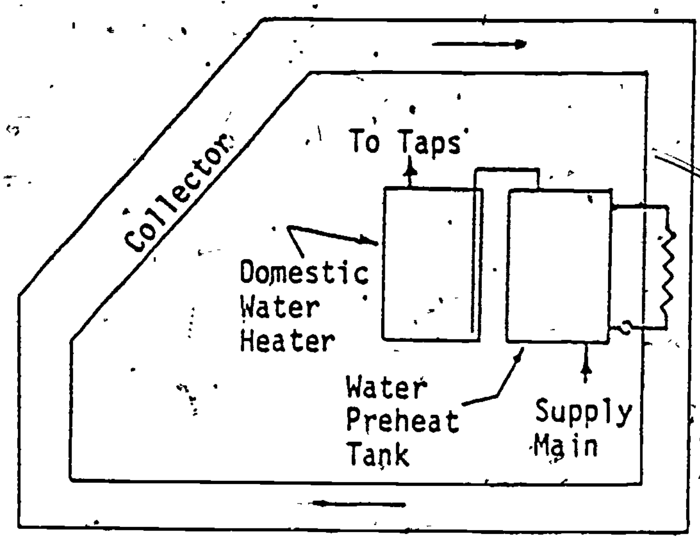
Collector
To Taps
Domestic
Water
Heater
Water
Preheat
Tank
Supply
Main

Figure 7-12(a). Air System Schematic for Service Hot Water.

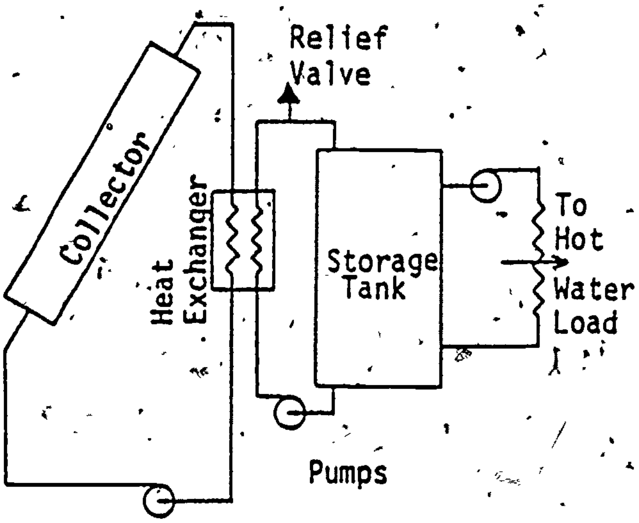
Relief
Valve
Collector
Heat
Exchanger
Storage
Tank
To
Hot
Water
Load
Pumps

Figure 7-12(b). Water System Schematic for Service Hot Water

The complete procedure has been automated on the SR-52, HP-65, and HP-67 programmable hand-held (Ref. 5, 6, 7) calculators. In addition, the procedure has also been programmed for a small digital computer. A sample output from the digital computer program is shown in Figure 7-13. This program, and similar programs, will be used during the design sessions of Modules 18 and 20.

INTERACTIVE PROGRAM

An interactive version of the design chart procedure that was discussed in the previous section has been developed at the University of Wisconsin. The program contains climatological and solar data for 112 locations in the United States and Canada. The program determines the performance of a solar system in any of these locations. The percentage of the load that will be supplied by solar for each month is determined. The user supplies the collector parameters necessary for the program; these are F_R' $(\tau\alpha)$ and F_R' U_L. The collector is modeled according to the equations presented in the earlier section. The user also specifies the slope and area of the collector array.

The building heating load is determined according to UA ΔT calculation or is specified, in monthly values, by the user. If the load is to be calculated, the user must provide a UA value in kJ/hr-^{o}C for the building. A service hot-water load may also be included. The user must specify the amount of service hot water (kg/day), the water set temperature (^{o}C), and the water main temperature (^{o}C). Internal heat generation from lights, people, etc., may also be specified in kJ/day. These factors are used in the program to determine the total load.

```
                    LATITUDE = 39.7              COLLECTOR TILT = 39.7        GROUND REFLECTIVITY = .20

      COLLECTOR AREA = 505.0 SQ. FT.                                  COVER TRANSMISSIVITY = .94
      PLATE ABSORBTIVITY = .90                                         COLL. EFF. FACTOR.FPRIME = .92
      OVERALL LOSS COEFF.UL= .97BTU/HR-SQ.FT.-F
      CAPACITY RATE OF COLL. FLUID = 7.0 BTU/HR-F-SQ.FT. COLL   CAPACITY RATE OF STORAGE FLUID = 8.2 BTU/HR-F-SQ.FT.COLL.
      EFFECTIVENESS OF COLL-TANK HX = .70                              HOUSE DESIGN = 24000.0 BTU/DD
      DOMESTIC USAGE PER DAY = 80.0 GAL/DAY                            DOMESTIC SUPPLY TEMP = 52.0 F
      DOMESTIC TANK TEMP = 140.0 F
      FUEL COST = 18.00 $/MBTU
      COST OF INSTALLATION = 16.00 $/SQ.FT. COLL.                      ANNUAL CAPITAL INVEST. CHARGE = 0.90
      STORAGE CAP. TO COLL. AREA RATIO = 15.4 LB WATER/SQ.FT. COLL   MATERIAL TANK COST = .120 $/LB STORAGE FLUID
      ADDITIONAL EQUIPMENT COSTS = 1400.

   MTH   SOLAR      MONTHLY    SOLAR      DEGREE    MONTHLY      MONTHLY        X         Y          PERCENT
         HORIZ      AVERAGE    TILT       DAYS      AVERAGE      TOTAL                               SOLAR
          BTU         R         BTU                 TEMP         LOAD
        ---------            ----------
        SQ.FT.-DAY           SQ.FT.-MTH

   1      741.0     1.81     41515.53    1132.0     28.48      28968480.      1.897      .503        .340
   2      988.0     1.52     42119.19     938.0     31.50      24138240.      2.022      .631        .420
   3     1478.0     1.26     57831.44     887.0     36.39      23088480.      2.278      .879        .594
   4     1696.0     1.03     52275.17     558.0     46.40      15134400.      3.171     1.212        .740
   5     1696.0      .90     47454.70     288.0     55.71       8712480.      5.372     1.912        .920
   6     1935.0      .85     49583.32      66.0     62.80       3326400.     12.998     5.232       1.000
   7     1917.0      .88     52034.29       6.0     64.81       1944480.     22.667     9.393       1.000
   8     1618.0      .97     48608.88       9.0     64.71       2016480.     21.872     8.462       1.000
   9     1519.0     1.16     52834.33     117.0     61.10       4550400.      9.610     4.076       1.000
  10     1143.0     1.44     50911.79     428.0     51.19      12072480.      3.989     1.480        .825
  11      818.0     1.74     42807.39     819.0     37.70      21398400.      2.360      .702        .468
  12      671.0     1.92     39989.86    1035.0     31.61      26640480.      2.028      .526        .351
            YEARLY DOLLAR SAVINGS = $756.63 WITH .547 OF THE REQUIRED LOAD CARRIED BY SOLAR
```

Figure 7-13. Sample Output

TABLE 7-2

SAMPLE OUTPUT FROM INTERACTIVE DESIGN PROGRAM

```
CODE                                                    VALUE      UNITS
 1      FRPRIME --TAU--ALPHA PRODUCT                      .695
 2      FRPRIME --UL PRODUCT                            16.900    KJ/hr-C-M^2
 3      BUILDING UA                                   1898.000    KJ/hr-C
 4      HOT WATER LOAD                                 300.000     KG/day
 5      WATER SET TEMPERATURE                           60.000       C
 6      WATER MAIN TEMPERATURE                          11.000       C
 7      COLLECTOR SLOPE                                 40.000     DEGREES
 8      CITY ABBREVIATION                                 IND
 9      COLLECTOR AREA                                  69.750    M^2=750 Ft^2
10      CONSTANT DAILY BLDG HEAT GENERATION              0.000     KJ/day
11      AIR SYSTEM = 1, WATER SYSTEM = 2                 2.000
12      PRINT OUT BY MONTH = 1, YEARLY = 2               1.000
TYPE YES IF YOU WISH TO CHANGE VALUE(S) IN LIST OF SYSTEM PARAMETERS
TYPE CHANGE IF YOU WISH TO CHANGE MONTHLY LOADS OR VALUES OF REFLECTIVITY
OTHERWISE TYPE READY OR STOP
READY
DO YOU WISH TO HAVE YOUR VALUES RELISTED YES=1, NO=2
2
INDIANAPO IN 39.44
TIME    PERCENT    INCIDENT     HEATING     WATER      DEGREE      AMBIENT
        SOLAR      SOLAR (MJ)   LOAD (MJ)   LOAD (MJ)  DAYS(C-day) TEMP.(C)
JAN      27.6        21.54        28.17       1.91       618.        -2.
FEB      40.1        24.66        24.02       1.72       527.        -0.
MAR      62.1        33.91        20.47       1.91       449.         4.
APR      88.8        34.79        10.93       1.85       240.        10.
MAY     100.0        39.89         4.48       1.91        98.        15.
JUN     100.0        40.64          .99       1.85        22.        18.
JUL     100.0        42.64         0.00       1.91         0.        18.
AUG     100.0        41.49         0.00       1.91         0.        18.
SEP     100.0        39.09         2.28       1.85        50.        17.
OCT      99.6        37.03         8.00       1.91       176.        13.
NOV      46.1        23.67        18.30       1.85       402.         5.
DEC      25.3        19.58        26.60       1.91       584.        -1.
YEAR     52.4       398.93       144.22      22.48      3166.
```

Finally, the user specifies whether the system being considered is an air or water system and whether the printout is to show monthly or yearly results. A sample output from this program with monthly results is shown in Table 7-2.

Economic considerations are also included in the program. The user has the option of having the program determine the optimal collector size. The optimum is determined on the basis of minimizing life-cycle costs.

SAMPLE DESIGN CURVES

The interactive program that was discussed in the previous section has been used to develop several design curves that may be used for collector sizing when using two specific collectors, one for air (SOLARON) and the other for water (PPG). These curves are shown in Figures 7-14 through 7-53. They were developed for the air and water systems in the ten locations indicated and tilt angles equal to the latitude and the latitude plus 15 degrees. They provide the annual fraction of load supplied by solar as a function of the house design heat load. For example, suppose we have a 15,000 Btu/DD house to be located in Boulder, Colorado. We see immediately from Figures 7-14 and 7-24 that a 500 Ft^2 SOLARON collector would provide between 70 and 75 percent of the annual heating load.

The detailed design methods presented in this module may be used to construct detailed design curves for any given collector type in any desired location. These curves may then be used in combination with an economic analysis to determine the final system design.

THE INTERACTIVE F-CHART PROGRAM

The interactive version of the f-chart program may be purchased from the University of Wisconsin - Madison by contacting Professor Beckman or Duffie at the Solar Energy Laboratory. We will illustrate its use and make it available during some of the computation sessions. The descriptions of variables and a program worksheet are given as follows:

Variable Description

1. Air System = 1, Liquid System = 2

The f-chart program can predict the performance of two types of solar domestic water and space heating systems. These "standard" systems use either air or a liquid as the transfer fluid and are illustrated in Figures 7-1 and 7-8. Solar air or liquid-based systems which heat domestic water exclusively can be modeled simply by inputting a space heating load of zero. [illegible] space heating only systems, input a water usage rate of zero.

2: Collector Area

The area of flat-plate collectors in your solar heating system.

3. $F_R'(\tau\alpha)_N$

If flat-plate collector experimental performance data are plotted as collector efficiency (η) vs $(T_{in}-T_{amb})/S$, a straight line can usually be fitted to the data points. Assuming that the data were taken with a specified fluid capacitance rate (liquid collector 210 KJ/hr-°C-m^2, air collector 45 KJ/hr-°C-m^2) and with all solar radiation at normal incidence to the plane of the collector, the intercept of the straight line with the η-axis is $F_R(\tau\alpha)_N$. F_R is the collector heat removal factor, $(\tau\alpha)_N$ is the transmittance-absorptance product of the collector cover system at normal incidence.

F_R' is a corrected F_R which in liquid-based systems accounts for the effect of the collector-to-storage heat exchanger. In air systems, $F_R = F_R'$ since there is no heat exchanger. F_R'/F_R can be calculated as follows:

$$F_R'/F_R = \left\{1 + \left[\frac{F_R U_L A}{(\dot{m}C_p)_c}\right] \left[\frac{(\dot{m}C_p)_c}{\epsilon_c(\dot{m}C_p)_{min}} - 1\right]\right\}^{-1}$$

$$F_R'(\tau\alpha)_N = F_R(\tau\alpha)_N \times F_R'/F_R$$

For nomenclature definitions, see "A Design Procedure for Solar Heating Systems" by S.A. Klein, W.A. Beckman and J.A. Duffie (Ref. 1).

4. $F_R'U_L$

$F_R U_L$ is the slope of the straight line η vs $(T_{in}-T_{amb})/S$ plot, if the plot is obtained as described above.

$$F_R'U_L = F_R U_L \times F_R'/F_R$$

5. Number of Transparent Covers

This refers to the number of glazings over the collector. The program assumes each cover is glass with an extinction coefficient-thickness product of .037. This information is used to calculate angle of incidence effects on the transmittance of glass.

6. Collector Slope

The angle between the plane of the collectors and horizontal.

7. Azimuth Angle

The angle between the horizontal projection of a ray normal to the plane of the collector, and due south. West is positive, east is negative.

8. Storage Capacity

The energy storage capacity of the storage unit in your solar system.

9. Effective Building UA

UA is calculated as the effective design space-heating load divided by the design temperature difference (indoor minus ambient). The effective design space-heating load should include infiltration and ventilation loads but should not take credit for heat generation within the space. The monthly space-heating load is then

LOAD = UA(deg-days/month)(hours/day)

10. Constant Daily Building Heat Generation

The user may take credit for heat generation within the space by typing in a daily generation rate here rather than including it in the building UA. This is actually a more correct way of including generation because UA should only include energy losses or gains which are dependent on ambient temperature.

Degree-days, used to estimate the load from UA, are based on a 65°F indoor temperature to partially account for heat generation. Consequently, credit for additional heat generation is usually not taken when calculating loads for residences.

11. Hot-Water Usage

If your solar system preheats water for domestic or process water use, this input is the average hot water usage rate.

12. Water Set Temperature

This is the temperature at which hot water must be supplied to your taps or to a process.

13. Water Main Temperature

This is the temperature at which water enters your system. It is usually the well water temperature or city mains water temperature.

14. City Call Number

The city call number identifies the location at which you want to make calculations. The first time you use FCHART it is recommended that you ask the program to give you a listing of available locations, all of which are numbered with a city call number. Keep this listing for future reference.

15. Print Out By Month = 1, By Year = 2

The results of the thermal analysis can be printed out by month or in yearly totals.

16. Economic Analysis? Yes = 1, No = 2

If desired, the program will perform a life-cycle cost economic analysis which compares the costs of the solar-assisted system with the costs of a conventional system on a present value basis. The program estimates the timing and amounts of annual cash flows using the following equations.

yearly cost with solar = mortgage payment + backup system fuel costs + misc. costs + property tax increase - income tax decrease with solar

yearly cost w/o solar = conventional system fuel cost - income tax decrease w/o solar

The terms are defined in the following. It is assumed that the solar backup system is identical to the conventional heating and domestic water system. Therefore only the additional investment due to the solar system need be considered in the analysis.

If an economic analysis is not desired, parameters 17 through 38 are ignored.

17. Use Optimized Collector Area = 1, Specified Area = 2

The user specifies a collector area via system parameter number 2. However, if an economically optimized collector area is desired, the program ignores the specified collector area and performs a numerical search for an optimum area. The criterion used is to find the collector area which minimizes the present value of all of the yearly costs of the solar-assisted system over the period of analysis.

18. Period of the Economic Analysis

This specifies the number of years over which the life-cycle cost analysis will be performed.

19. Collector Area Dependent Costs

Some of the extra costs of a solar heating system above the conventional system are collector area dependent. These include the costs of storage and of the collector.

20. Constant Solar Costs

This refers to extra costs of solar heating systems above the conventional system which are not dependent on the collector size. Examples are costs for architectural modifications, piping or ducts, controls, and pumps or blowers.

21. Down Payment (% of Original Investment)

The original investment refers to the extra investment required to put in the solar system. Therefore the % which is paid down on the solar system equals the ratio of the incremental increase in the down payment required by the lender to the incremental increase in the size of the loan required due to the solar system.

22. Annual Interest Rate on Mortgage

This is the annual interest rate charged by your lender.

23. Term of the Mortgage

The number of years over which you must pay off the loan.

24. Annual Nominal (Market) Discount Rate

This refers to the annual rate of return which you make with your money in your best investment opportunity. The annual nominal or market rate of return equals the real rate of return plus the general inflation rate. For the typical homeowner the real rate of return is 1-2%; for business, 3-4%.

25. Expenses (Insurance, Maintenance) of System in First Year

All additional yearly expenses due to the solar system which cannot be input anywhere else should be included here.

26. Annual Increase in Above Expenses

Allowance can be made for the annual rate of increase of insurance and maintenance costs (i.e. the general inflation rate) via this parameter.

27. Present Cost of Auxiliary Fuel (CF)

This is the actual present cost of the backup system fuel, times 100, divided by the efficiency of the backup system heating unit. The actual present cost of the fuel should include any fuel adjustment charges beyond the standard rate.

28. CF Rise: Linear = 1, %/yr. = 2, Seq. of Values = 3

The program user may allow fuel costs to rise in any of three possible ways so that any scenario can be investigated. These increases should include general inflation plus any net increases in fuel costs.

29. If 1, What is the Slope of CF Increase?

If a linear fuel cost rise is assumed, the slope of increase is required. Otherwise this parameter is ignored.

30. If 2, What is the Annual Rate of CF Rise?

If a %/yr. fuel cost rise is desired, the annual rate of fuel cost increase must be input here. Otherwise this parameter is ignored.

31. Economic Print Out by Year = 1, Cumulative = 2

If a yearly print out is desired, several cash flows are printed each year of the economic analysis. If a cumulative print out is desired, the present value of the yearly costs over the period are output for the building with and without a solar energy system.

32. Effective Federal-State Income Tax Rate

State income taxes paid are deductible on federal returns; therefore the effective federal-state income tax rate is calculated as

$$\text{Effective Rate} = \text{Federal Rate} + \text{State Rate} - (\text{Federal Rate}) \times (\text{State Rate})$$

33. True Property Tax Rate per $ of Original Investment

Property tax rates are applied to your assessed value. Therefore an estimate of assessed value as a percentage of original investment is required so that

$$\frac{\text{Tax Rate}}{\$\ \text{Original Invest.}} = \left(\frac{\text{Tax Rate}}{\text{Assessed Value}}\right) \times \left(\frac{\text{Assessed Value}}{\text{Original Invest.}}\right)$$

34. Income Producing Building? Yes = 1, No = 2

The economic analyses for commercial and residential buildings are different because businesses benefit from more income tax deductions due to the added investment required for the solar heating system. For the homeowner, interest and property taxes paid are deductible on income taxes. For a business, interest, depreciation, fuel expenses, property taxes and maintenance and insurance costs are all deductible.

If your building is not income producing and does not qualify as a business investment, parameters 35 through 38 are ignored.

The yearly cost equations are given in the discussion below system parameter 16. For a non-income-producing building, such as a residence, the income tax terms are given below.

$$\substack{\text{income tax}\\ \text{decrease}\\ \text{with solar}} = \substack{\text{tax}\\ \text{rate}} \times \{\text{interest paid} + \text{property tax paid}\}$$

$$\substack{\text{income tax}\\ \text{decrease}\\ \text{w/o solar}} = 0.$$

However, for a commercial building the income tax terms are as follows:

$$\substack{\text{income tax}\\ \text{decrease}\\ \text{with solar}} = \substack{\text{tax}\\ \text{rate}} \times \{\substack{\text{interest}\\ \text{paid}} + \substack{\text{property}\\ \text{tax}\\ \text{paid}} + \substack{\text{misc.}\\ \text{expense}} + \substack{\text{backup sys.}\\ \text{fuel cost}} + \text{depreciation}\}$$

$$\substack{\text{income tax}\\ \text{decrease}\\ \text{w/o solar}} = \text{tax rate} \times \{\text{conventional system fuel costs}\}$$

Although commercial building owners have more income tax deductions due to the solar investment than homeowners, they do not save as much in fuel costs since fuel costs are deductible whether or not they install a solar heating system.

Keep in mind that the interest, property tax, miscellaneous expense, and depreciation deductions refer only to the incremental increase in these deductions due to the solar investment. Therefore these terms do not appear in the "without solar" income tax decrease equation.

35. Dprc.: Straight Line = 1, Declining Balance = 2, Sum-of-Years - Digits = 3, None = 4

Any of the standard methods of depreciation can be used. Depreciation deductions due to the extra investment due to solar are calculated in order to estimate the income tax savings.

36. If 2, What % of Straight Line Depreciation Rate is Desired?

The federal government allows several rates at which investments can be written off using the declining balance method. These rates are expressed as the % of straight line depreciation rate allowed.

37. Useful Life for Depreciation Purposes

This is the length of time over which you intend to depreciate out your investment.

38. Salvage Value at End of Depreciation Period

An estimate of the system's salvage value at the end of the depreciation period is required in order to calculate depreciation.

F-CHART PROGRAM WORKSHEET

CODE	VARIABLE DESCRIPTION	SI	UNITS	ENGLISH	UNITS
1	AIR SYSTEM=1,LIQUID SYSTEM=2				
2	COLLECTOR AREA		M2		FT2
3	FRPRIME-TAU-ALPHA PRODUCT(NORMAL INCIDENCE)				
4	FRPRIME-UL PRODUCT		KJ/HR-C-M2		BTU/H-F--
5	NUMBER OF TRANSPARENT COVERS				
6	COLLECTOR SLOPE		DEGREES		DEGREES
7	AZIMUTH ANGLE (E.G. SOUTH=0, WEST=90)		DEGREES		DEGREES
8	STORAGE CAPACITY		KJ/C-M2		BTU/F-FT2
9	EFFECTIVE BUILDING UA		KJ/HR-C		BTU/HR-F
10	CONSTANT DAILY BLDG HEAT GENERATION		KJ/DAY		BTU/DAY
11	HOT WATER USAGE		L/DAY		GAL/DAY
12	WATER SET TEMPERATURE		C		F
13	WATER MAIN TEMPERATURE		C		F
14	CITY CALL NUMBER				
15	PRINT OUT BY MONTH=1, BY YEAR=2				
16	ECONOMIC ANALYSIS ? YES=1, NO=2				
17	USE OPTMZD. COLLECTOR AREA=1, SPECFD. AREA=2.				
18	PERIOD OF THE ECONOMIC ANALYSIS		YEARS		YEARS
19	COLLECTOR AREA DEPENDENT SYSTEM COSTS		$/M2 COLL		$/FT2 COLL
20	CONSTANT SOLAR COSTS		$		$
21	DOWN PAYMENT(% OF ORIGINAL INVESTMENT)		%		%
22	ANNUAL INTEREST RATE ON MORTGAGE				
23	TERM OF MORTGAGE		YEARS		YEARS
24	ANNUAL NOMINAL(MARKET) DISCOUNT RATE		%		%
25	EXPENSES(INSUR.,MAINT.) OF SYSTEM IN 1ST YEAR		$		$
26	ANNUAL % INCREASE IN ABOVE EXPENSES		%		%
27	PRESENT COST OF AUXILIARY FUEL (CF)		$/GJ		$/MBTU
28	CF RISE: LINEAR=1,%/YR=2,SEQ. OF VALUES=3				
29	IF 1, WHAT IS THE SLOPE OF CF INCREASE?		$/GJ-YR		$/MBTU-
30	IF 2, WHAT IS THE ANNUAL RATE OF CF RISE		%		%
31	ECONOMIC PRINT OUT BY YEAR=1, CUMULATIVE=2				
32	EFFECTIVE FEDERAL-STATE INCOME TAX RATE		%		%
33	TRUE PROP. TAX RATE PER $ OF ORGINAL INVEST.		%		%
34	INCOME PRODUCING BUILDING? YES=1,NO=2				
35	DPRC.: STR.LN=1,DC.BAL.=2,SM-YR-DGT=3,NONE=4.				
36	IF 2, WHAT % OF STR.LN DPRC.RT.IS DESIRED?		%		%
37	USEFUL LIFE FOR DEPREC. PURPOSES		YEARS		YEARS
38	SALVAGE VALUE AT END OF DEPREC. PERIOD		$		$

REFERENCES

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3. G. Löf, R. A. Tybout, "Solar House Heating", Natural Resources Journal, April 1970.
4. S. Klein, W. Beckman, J. Duffie, "A Design Procedure for Solar Air Heating Systems", International Solar Energy Society Conference, Winnipeg, Manitoba, Canada, August 1976.
5. C. B. Winn, G. R. Johnson, "Solar Energy Analysis Programs for Programmable Handheld Calculators, Texas Instruments SR-52 Edition", Report TR-99, SEEC, Inc., Fort Collins, Colorado, 80522.
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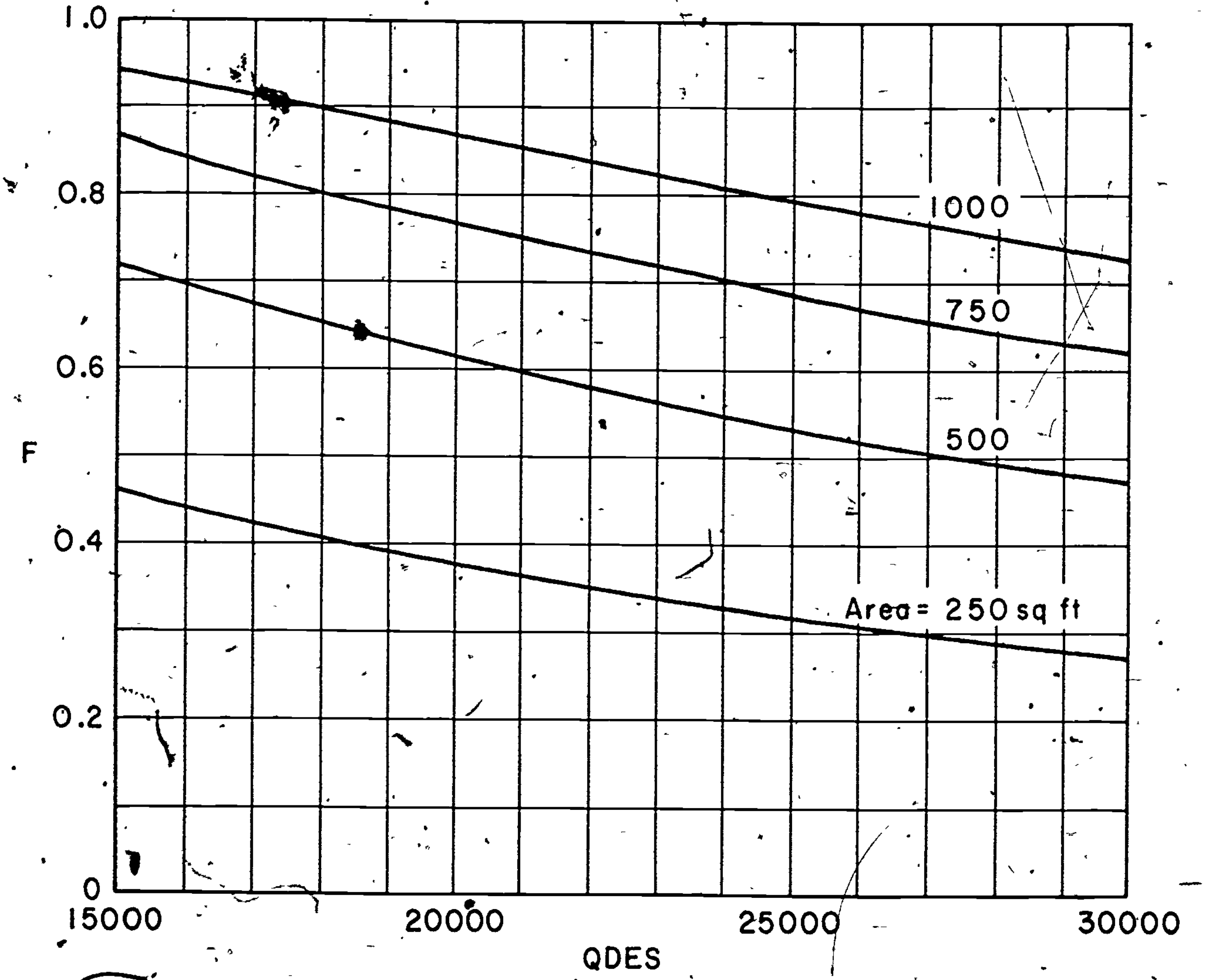

Figure 7-14.

AIR SYSTEM
SOLARON COLLECTOR
SLOPE = LAT
LOCATION: BOULDER

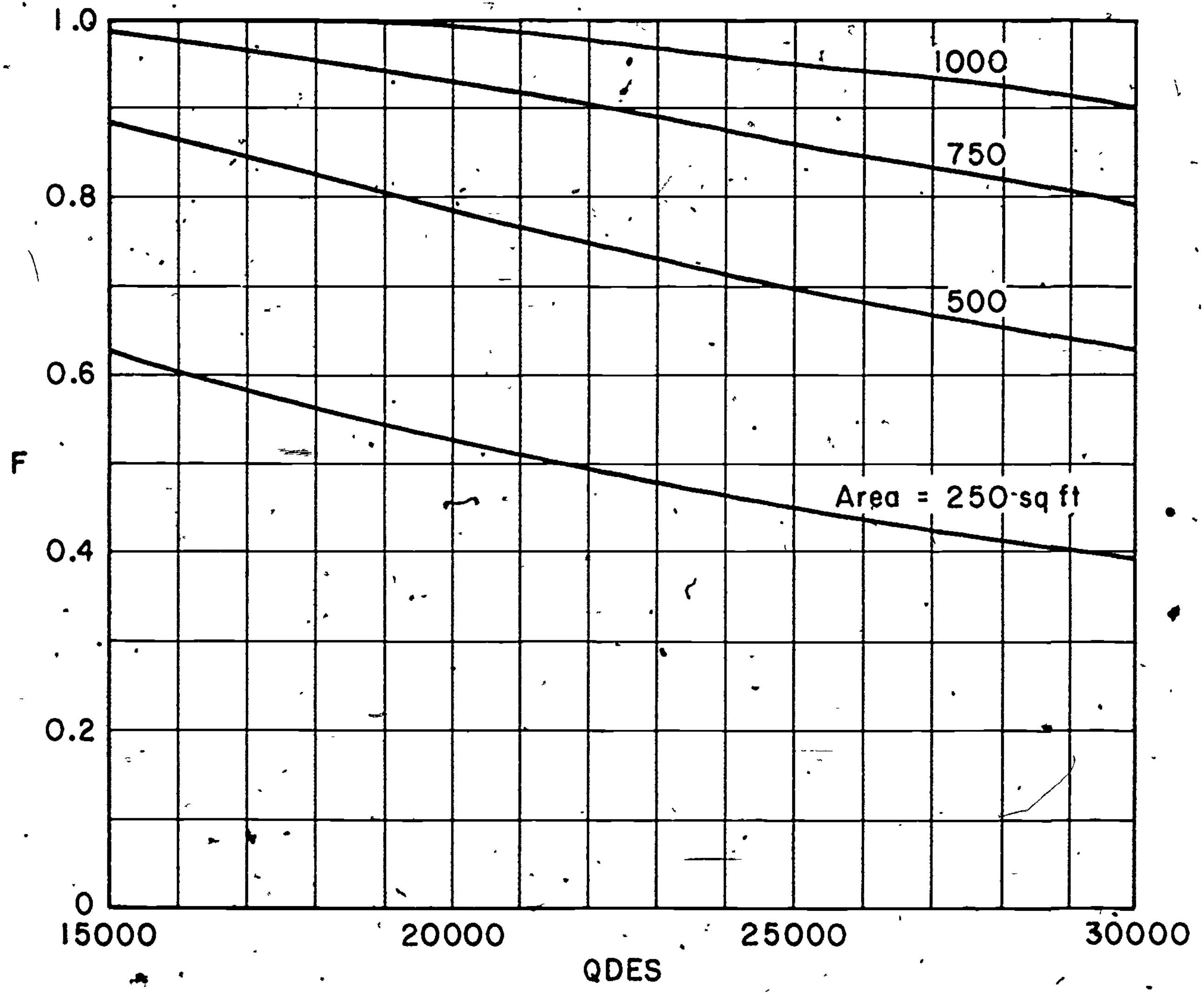

Figure 7-15.

AIR SYSTEM
SOLARON COLLECTOR
SLOPE = LAT
LOCATION: ALBUQUERQUE

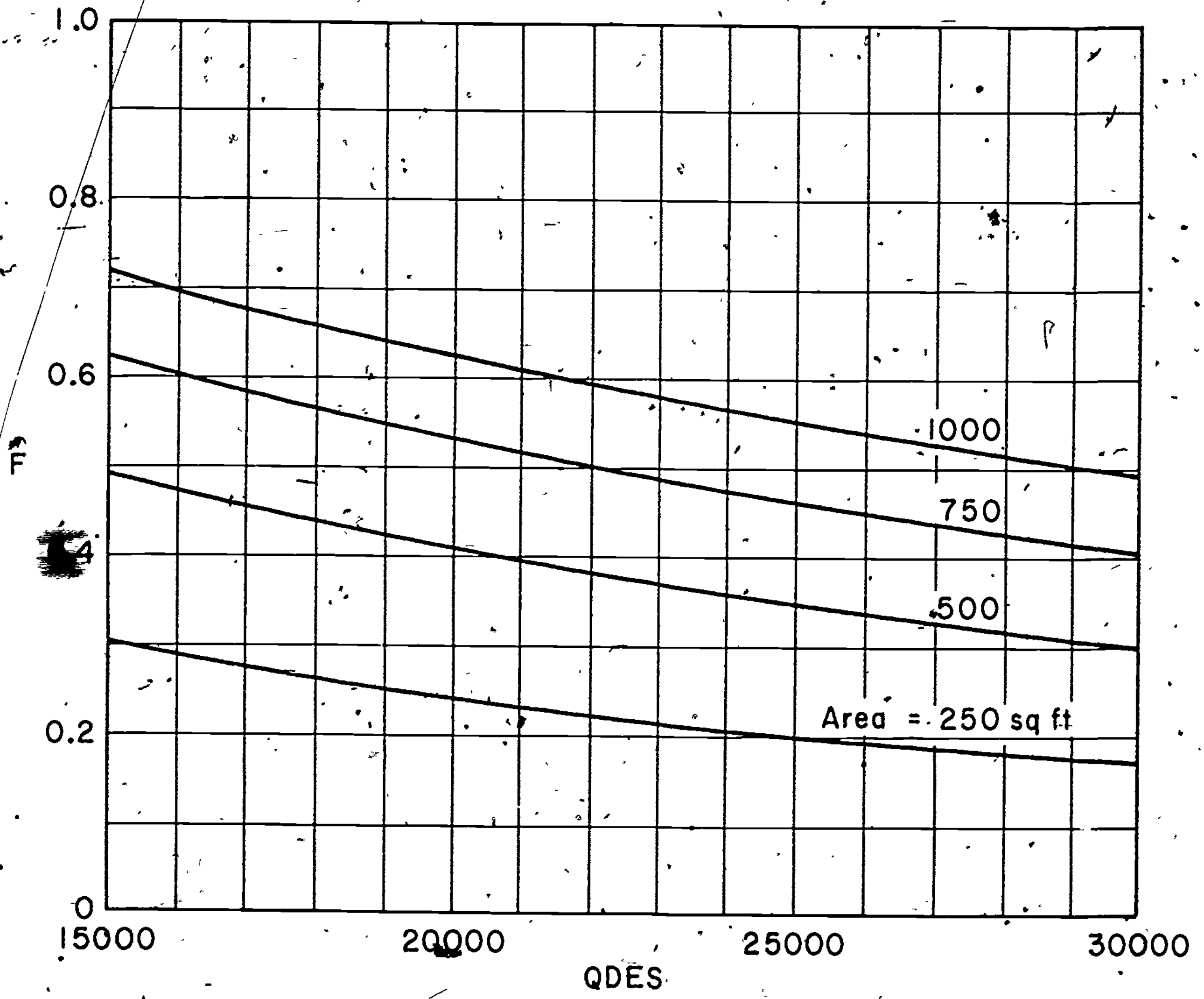

Figure 7-16.

AIR SYSTEM
SOLARON COLLECTOR
SLOPE = LAT
LOCATION: MADISON

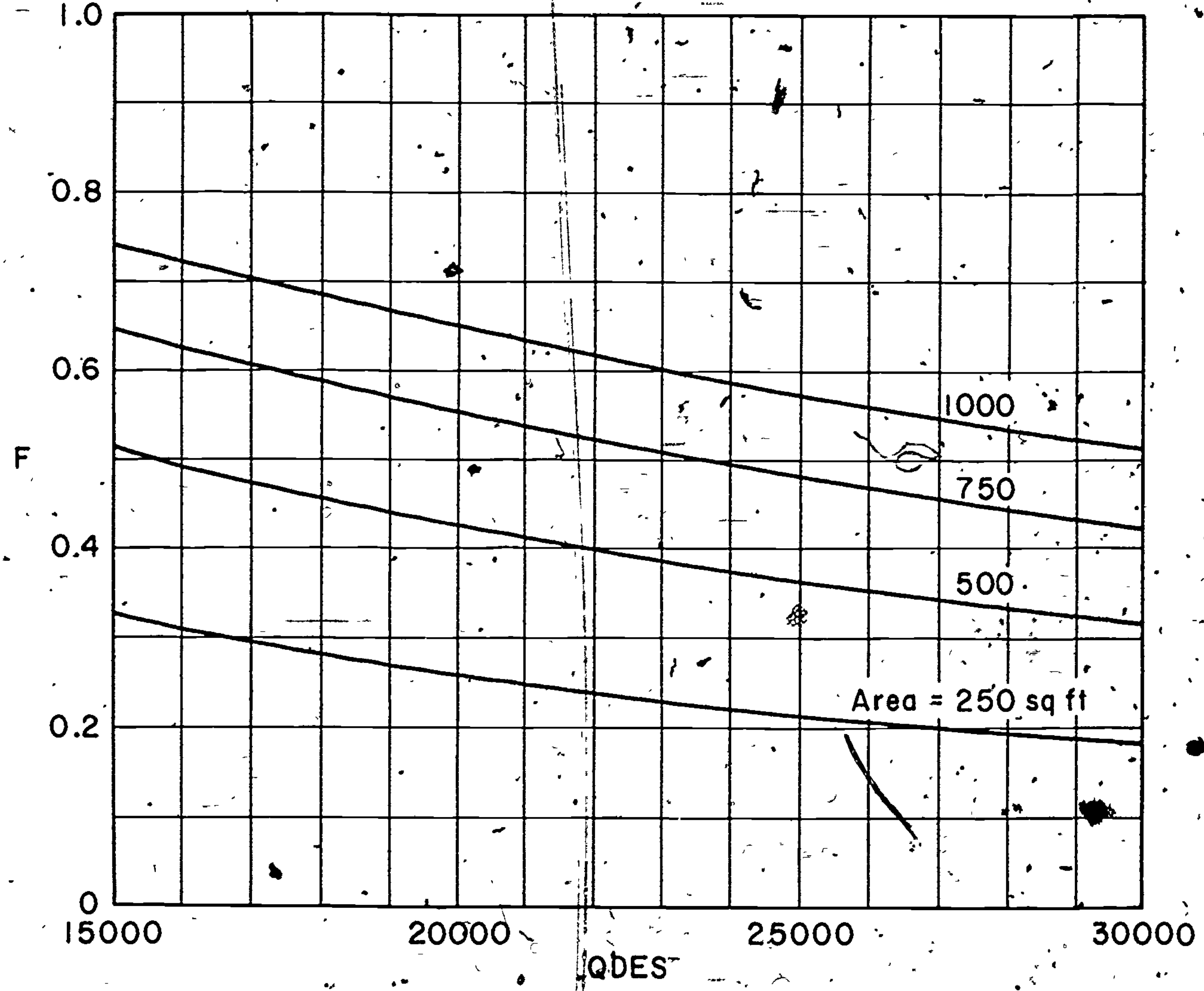

Figure 7-17.

AIR SYSTEM
SOLARON COLLECTOR
SLOPE = LAT
LOCATION: BOSTON

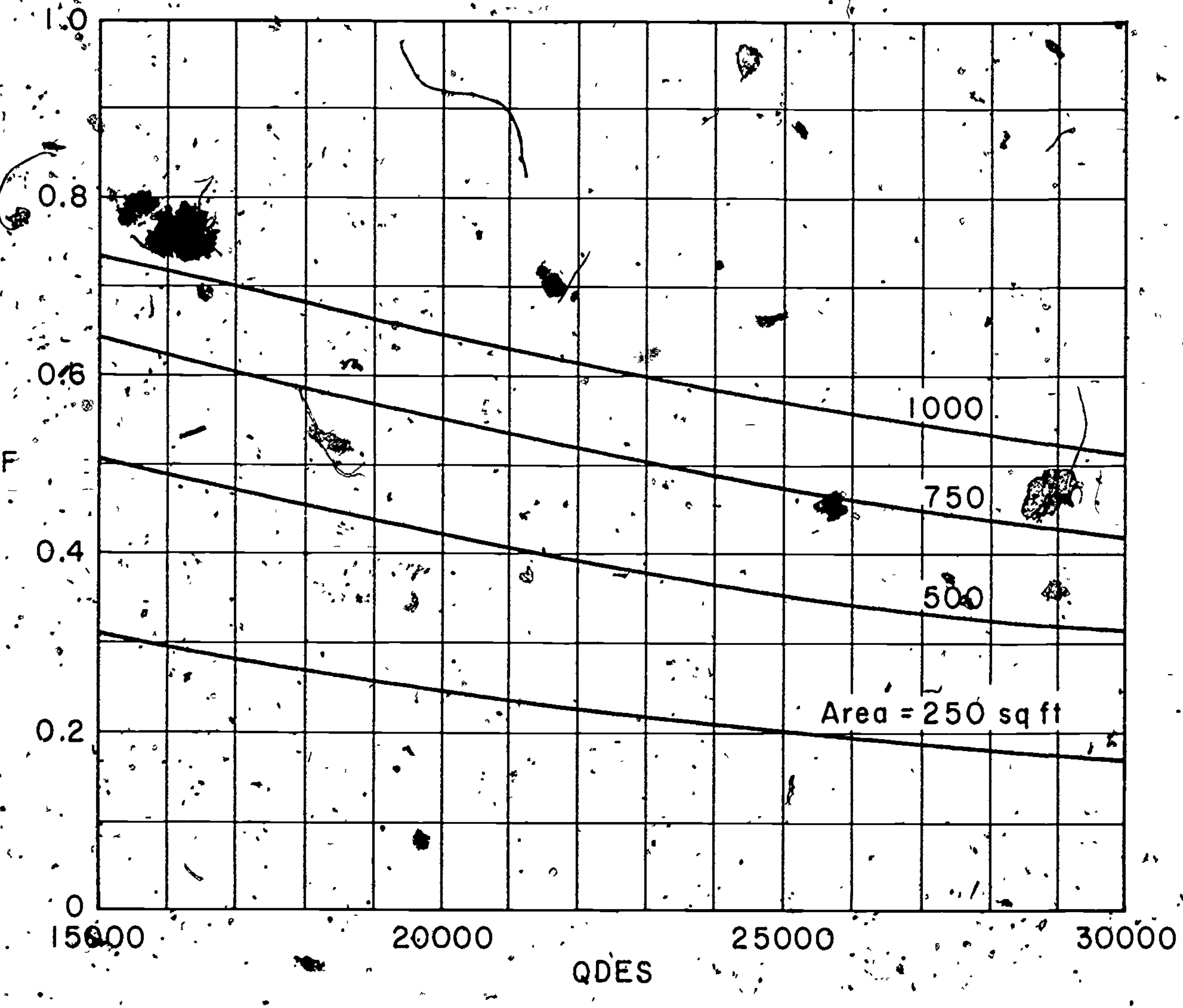

Figure 7-18.

AIR SYSTEM
SOLARON COLLECTOR
SLOPE = LAT
LOCATION: ALBANY

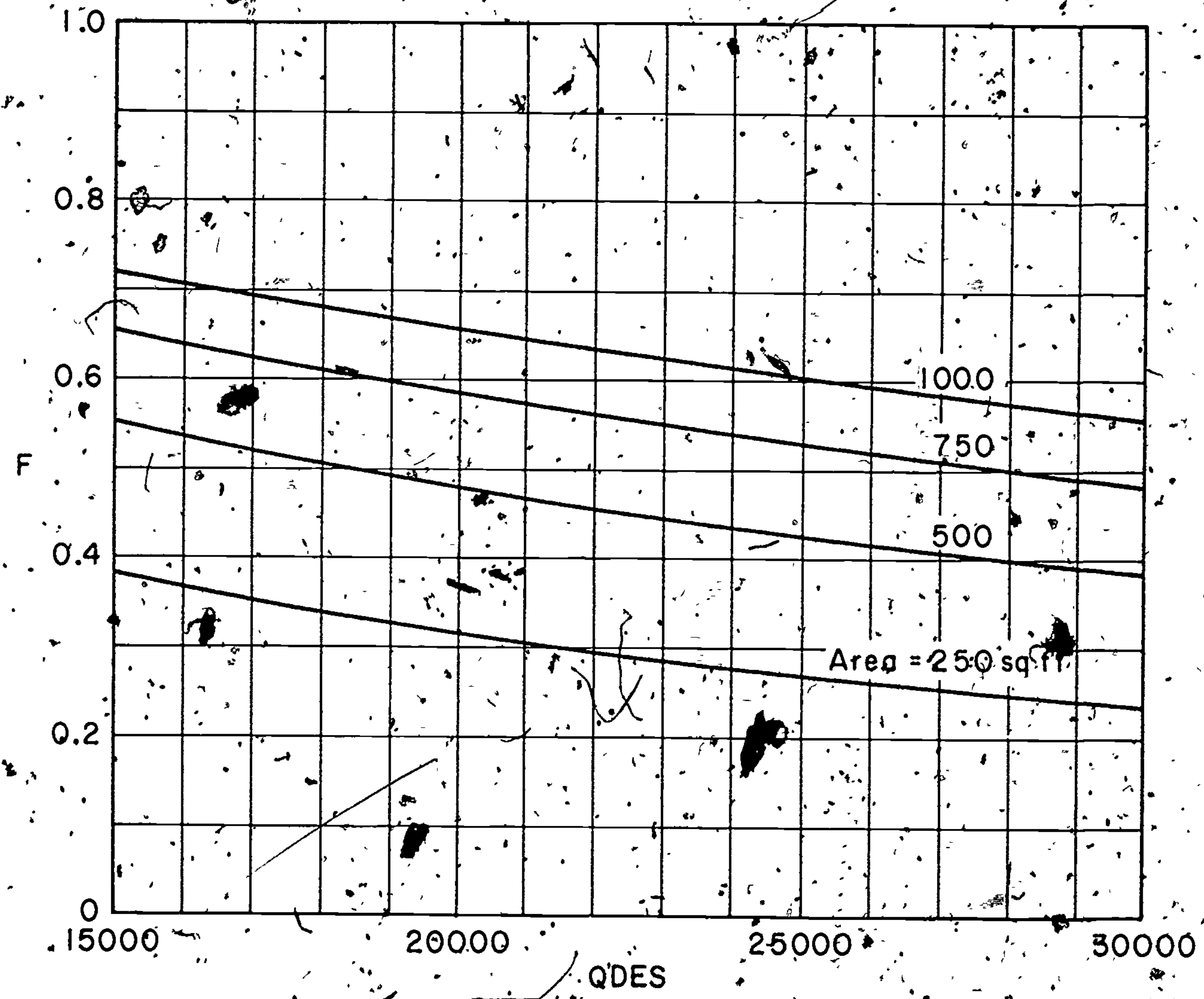

Figure 7-19.

AIR SYSTEM
SOLARON COLLECTOR
SLOPE = LAT
LOCATION: SEATTLE

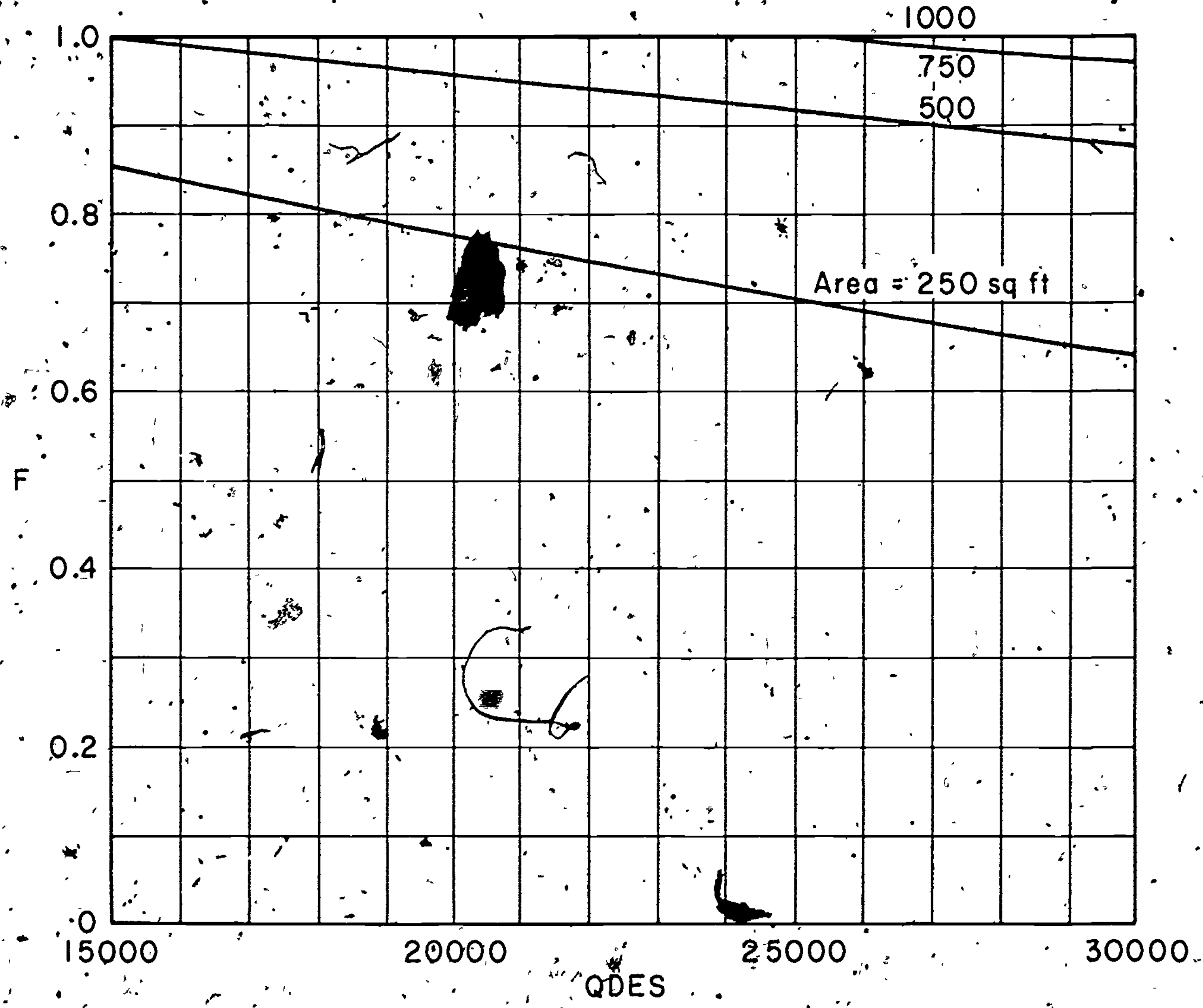

Figure 7-20.

AIR SYSTEM
SOLARON COLLECTOR
SLOPE = LAT
LOCATION: GAINESVILLE

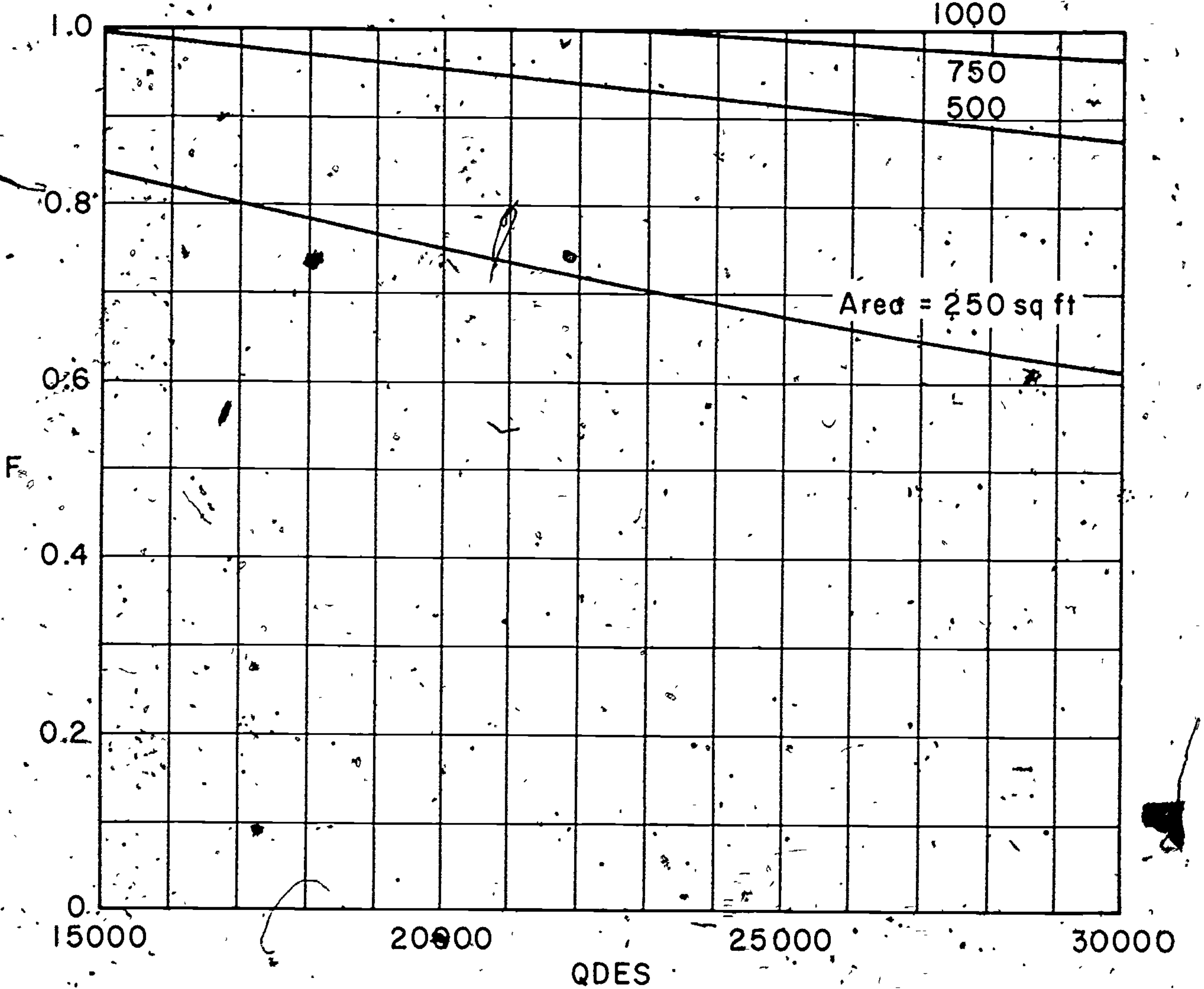

Figure 7-21.

AIR SYSTEM
SOLARON COLLECTOR
SLOPE = LAT
LOCATION: SANTA MARIA

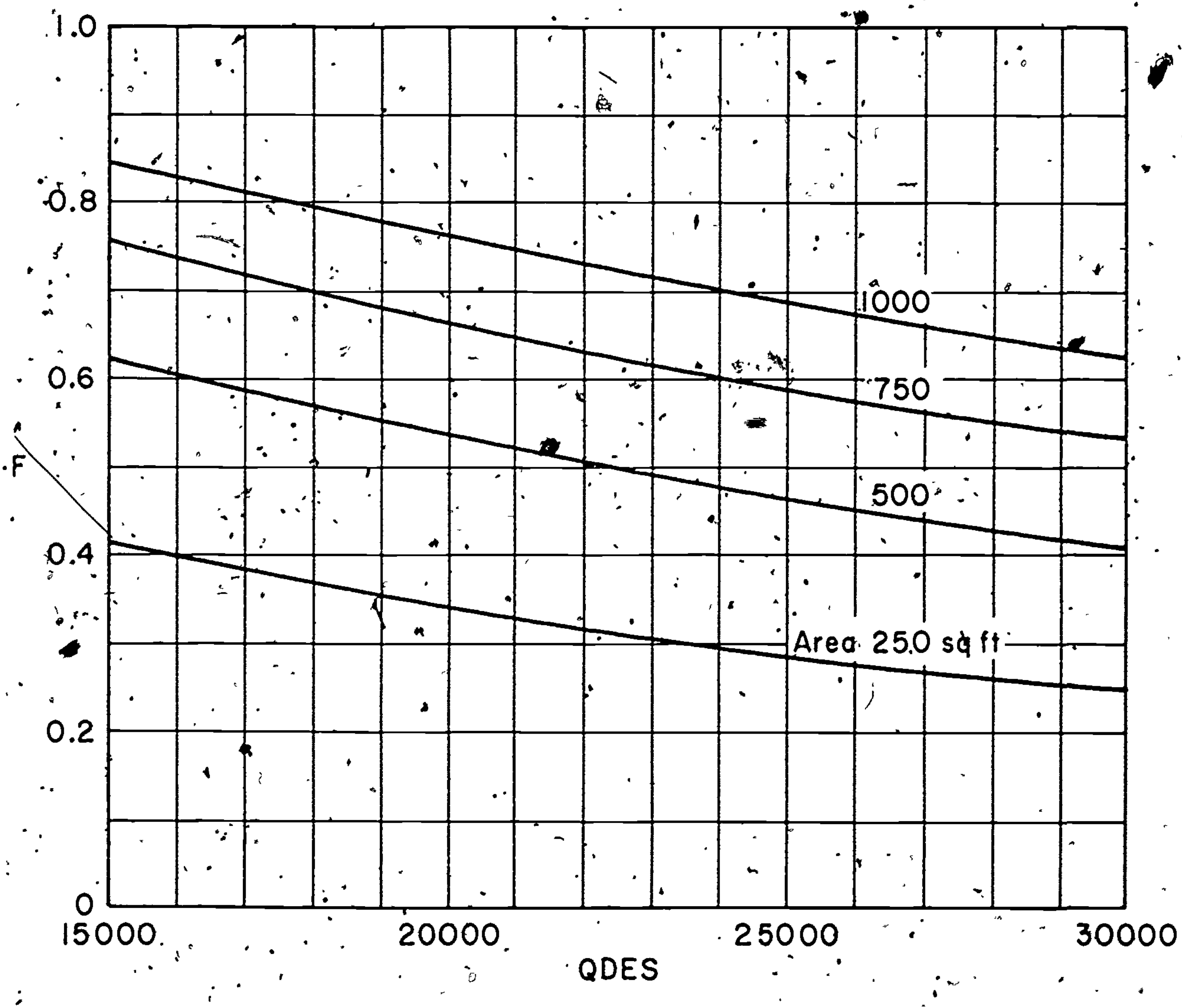

Figure 7-22.

AIR SYSTEM
SOLARON COLLECTOR
SLOPE = LAT
LOCATION: WASH., D. C.

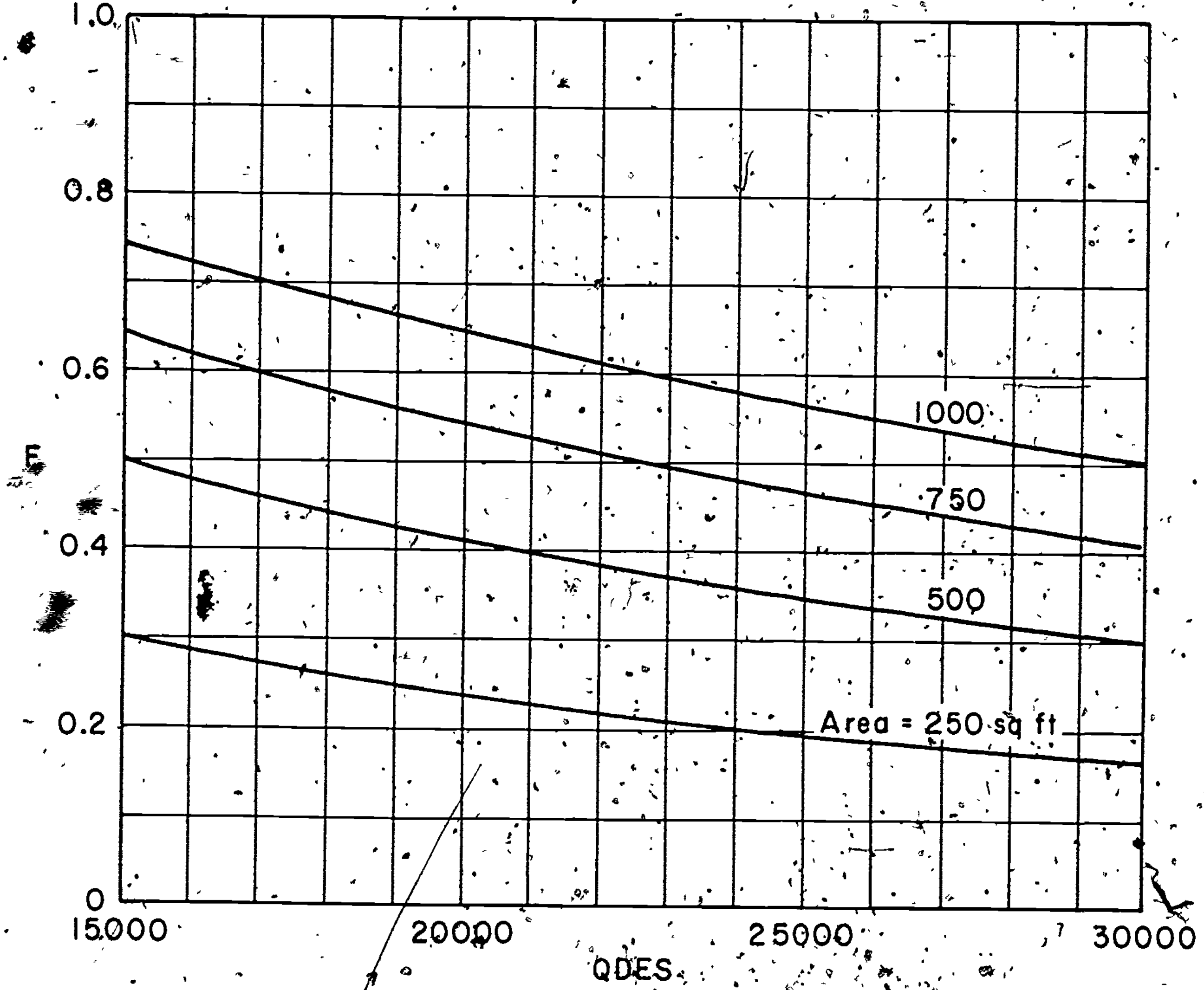

Figure 7-23

AIR SYSTEM
SOLARON COLLECTOR
SLOPE = LAT
LOCATION: ST. CLOUD

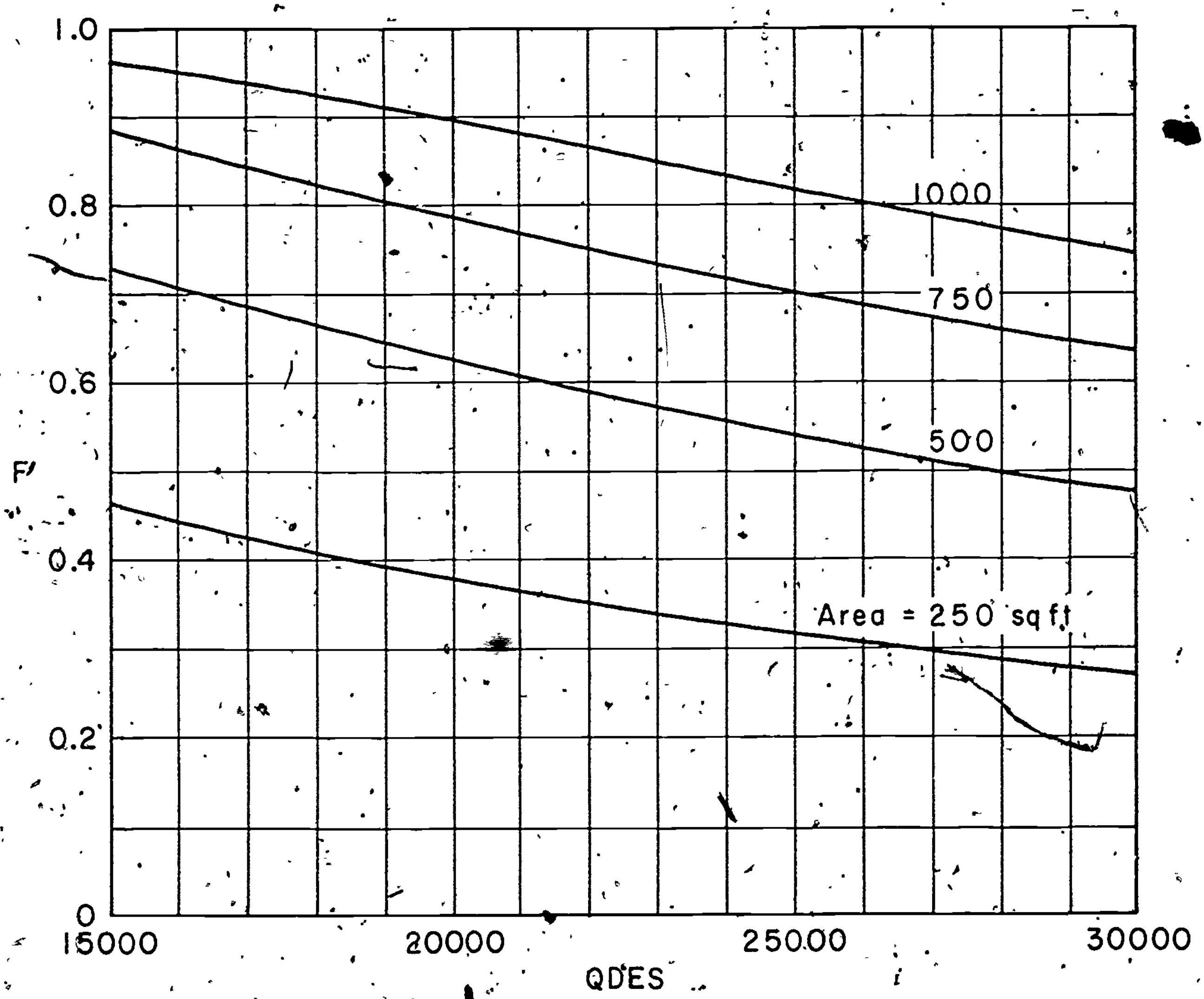

Figure 7-24.

AIR SYSTEM
SOLARON COLLECTOR
SLOPE = LAT + 15
LOCATION: BOULDER

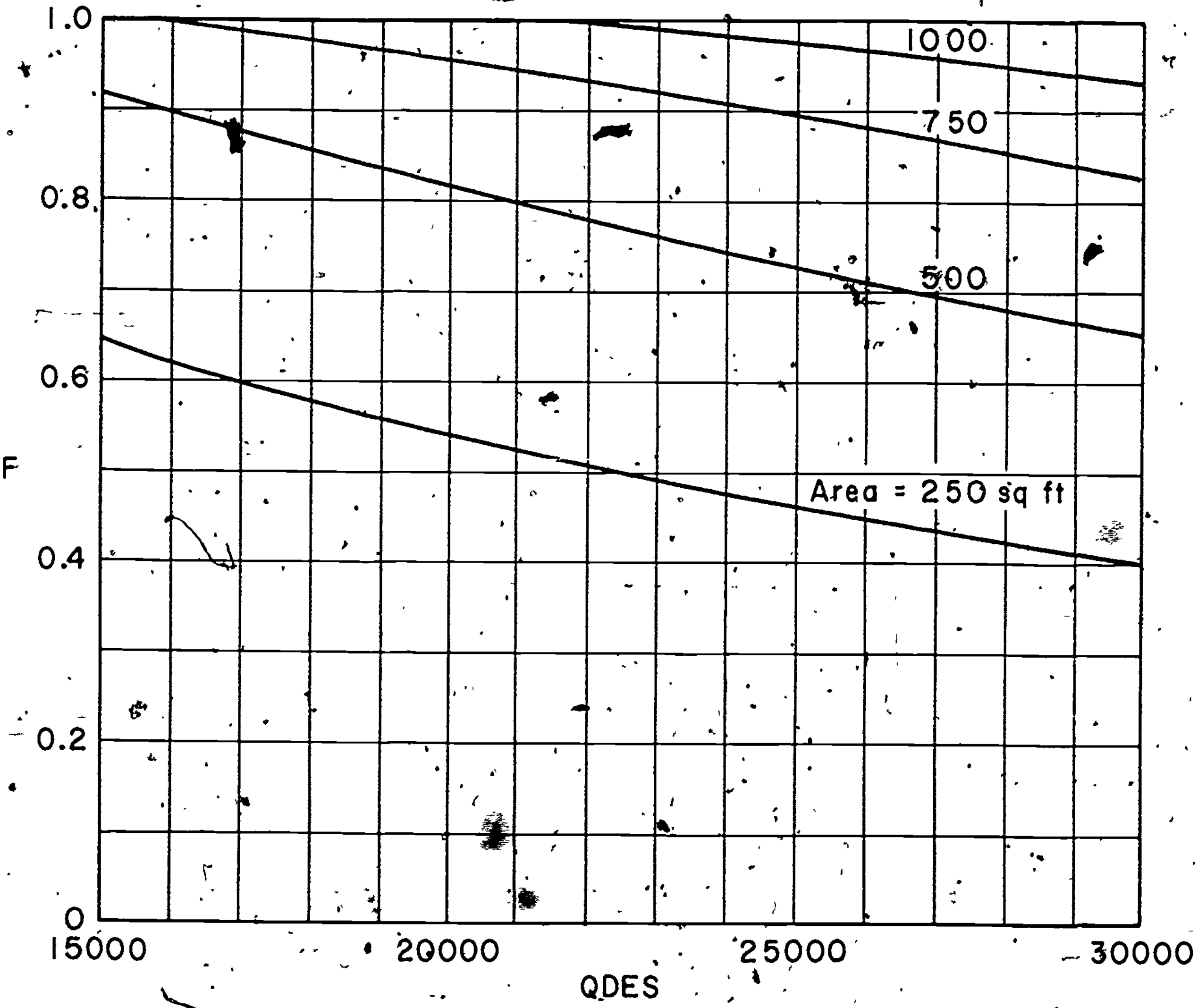

Figure 7-25.

AIR SYSTEM
SOLARON COLLECTOR
SLOPE = LAT + 15
LOCATION: ALBUQUERQUE

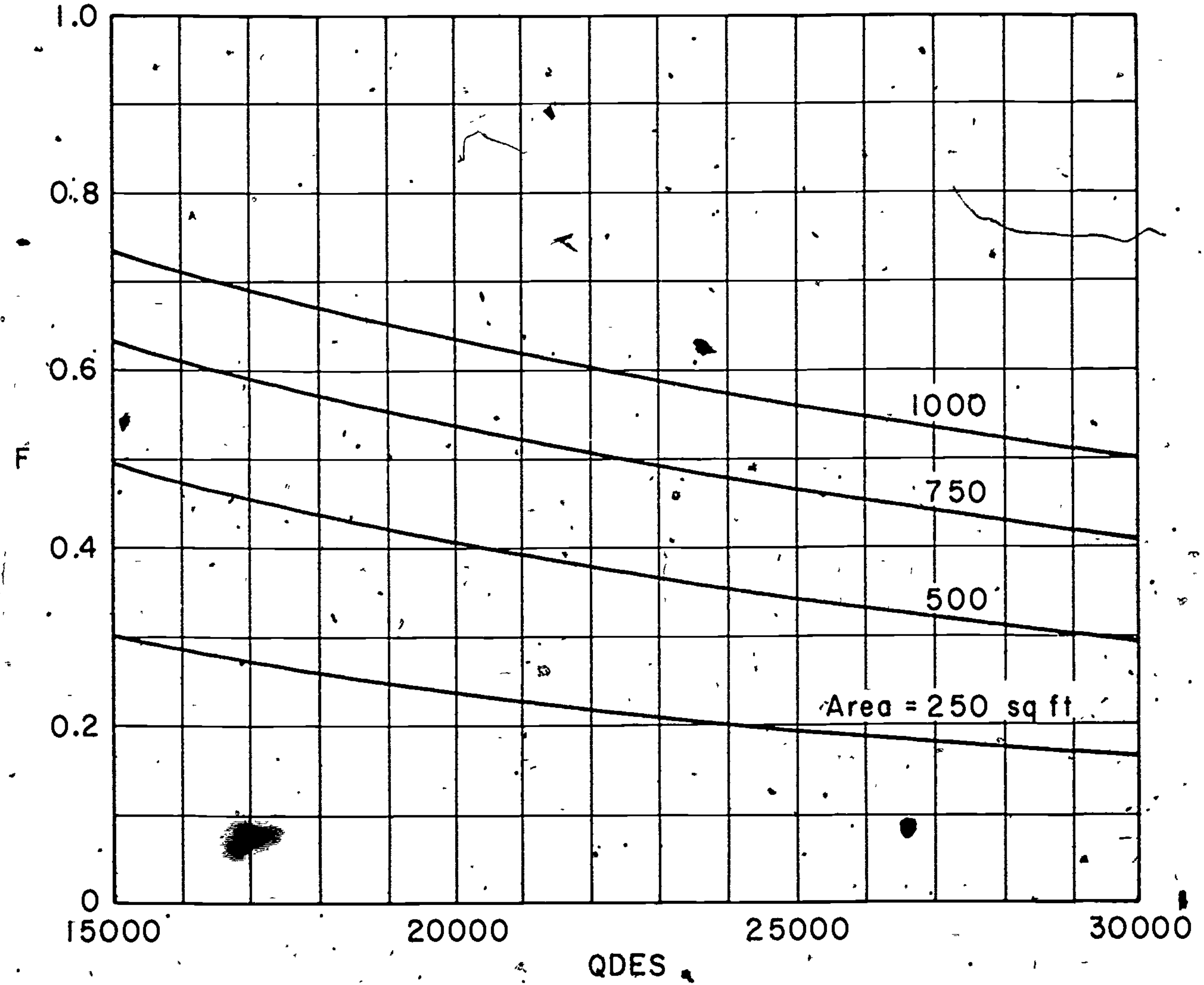

Figure 7-26.

AIR SYSTEM
SOLARON COLLECTOR
SLOPE = LAT + 15
LOCATION: MADISON

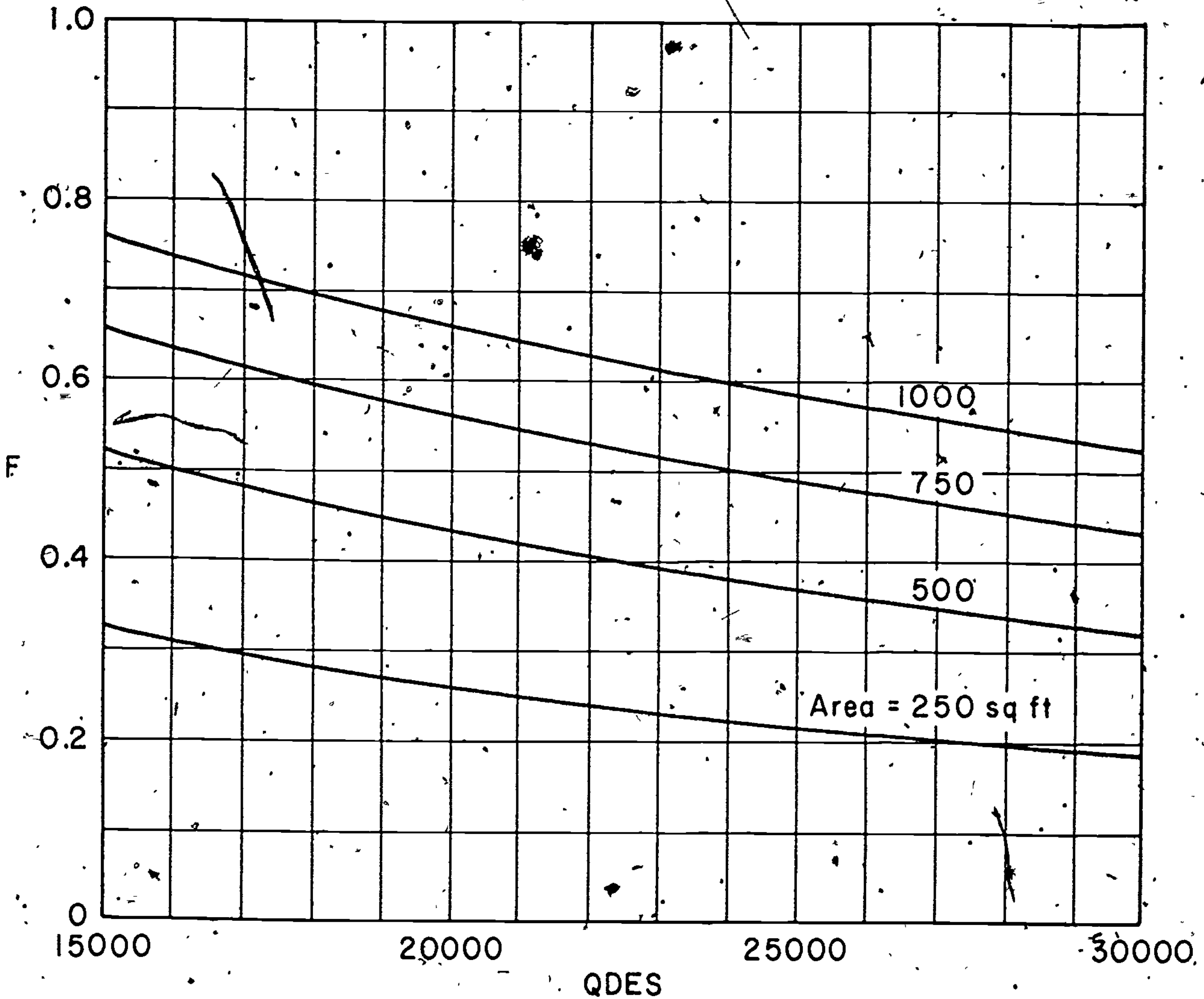

Figure 7-27.

AIR SYSTEM
SOLARON COLLECTOR
SLOPE = LAT + 15
LOCATION: BOSTON

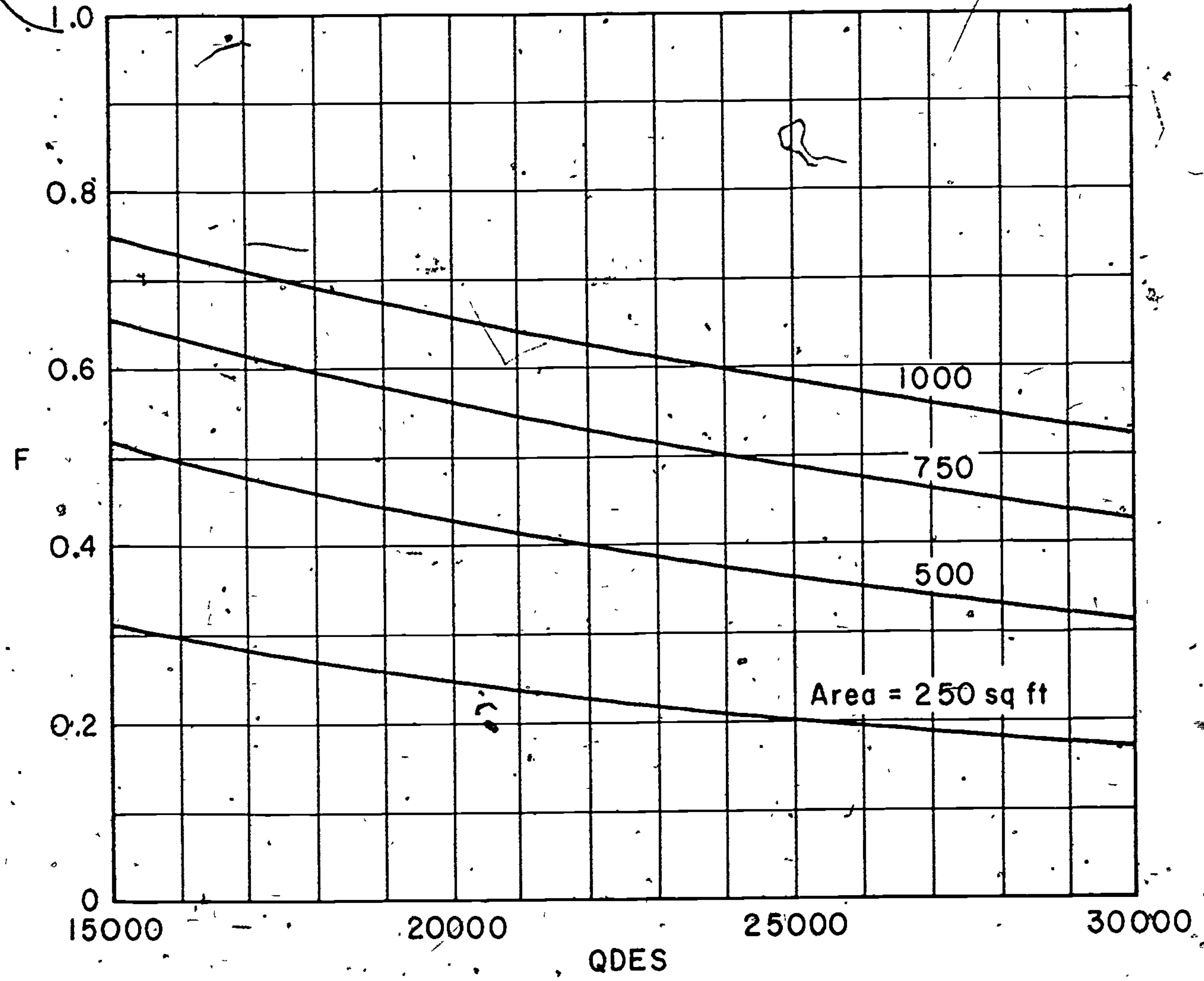

Figure 7-28.

AIR SYSTEM
SOLARON COLLECTOR
SLOPE = LAT + 15
LOCATION: ALBANY

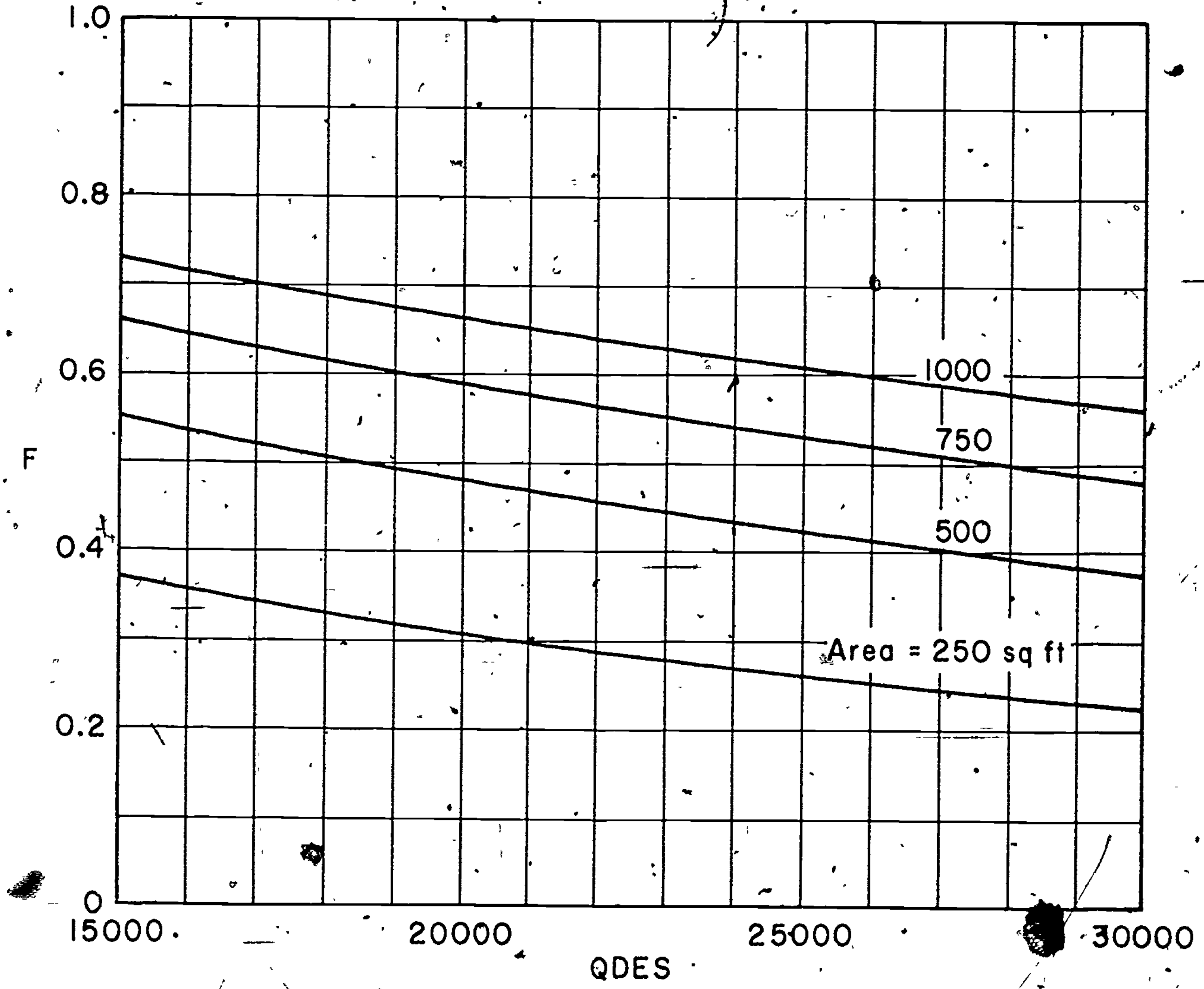

Figure 7-29.

AIR SYSTEM
SOLARON COLLECTOR
SLOPE = LAT + 15
LOCATION: SEATTLE

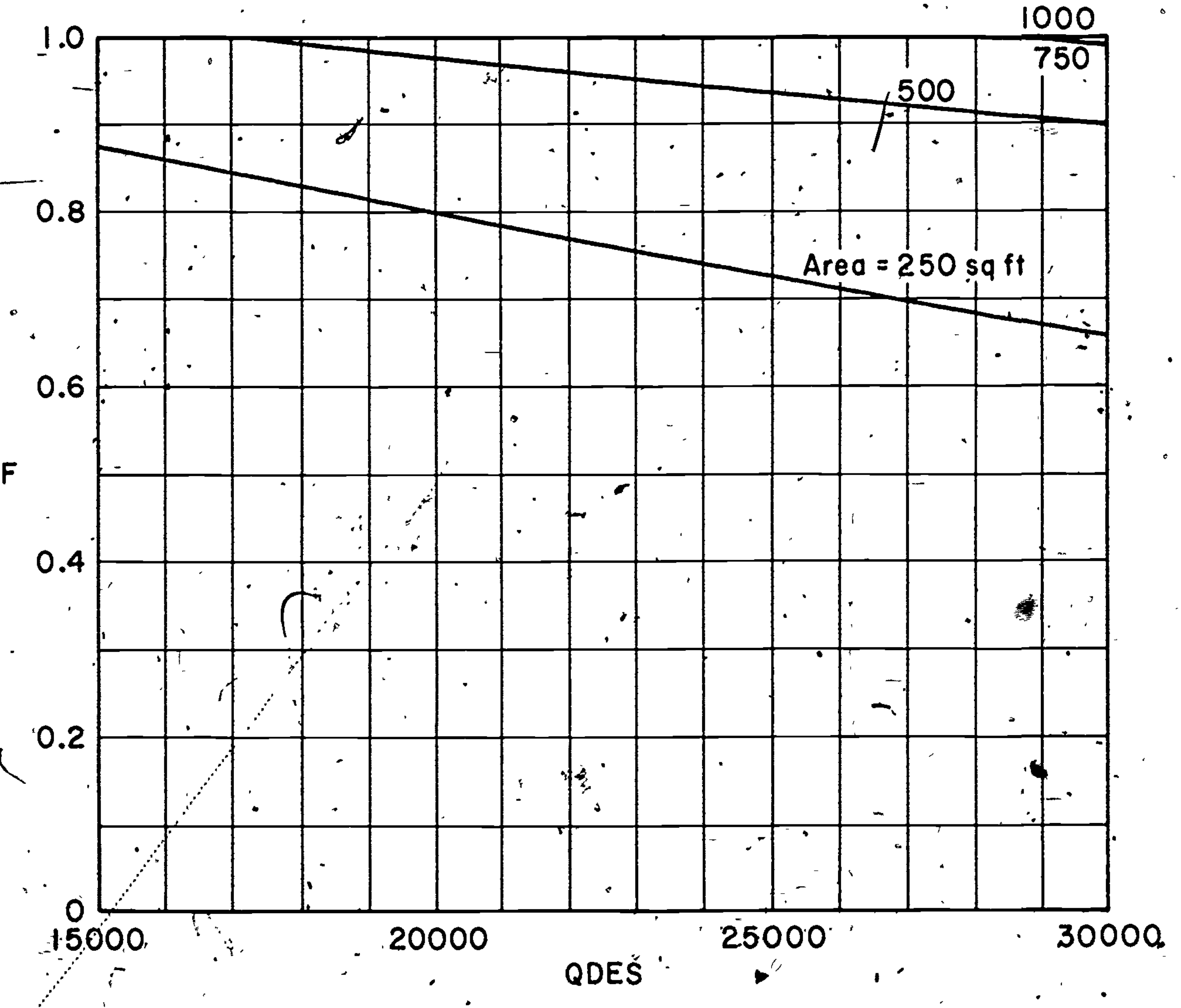

Figure 7-30.

AIR SYSTEM
SOLARON COLLECTOR
SLOPE = LAT + 15
LOCATION: GAINESVILLE

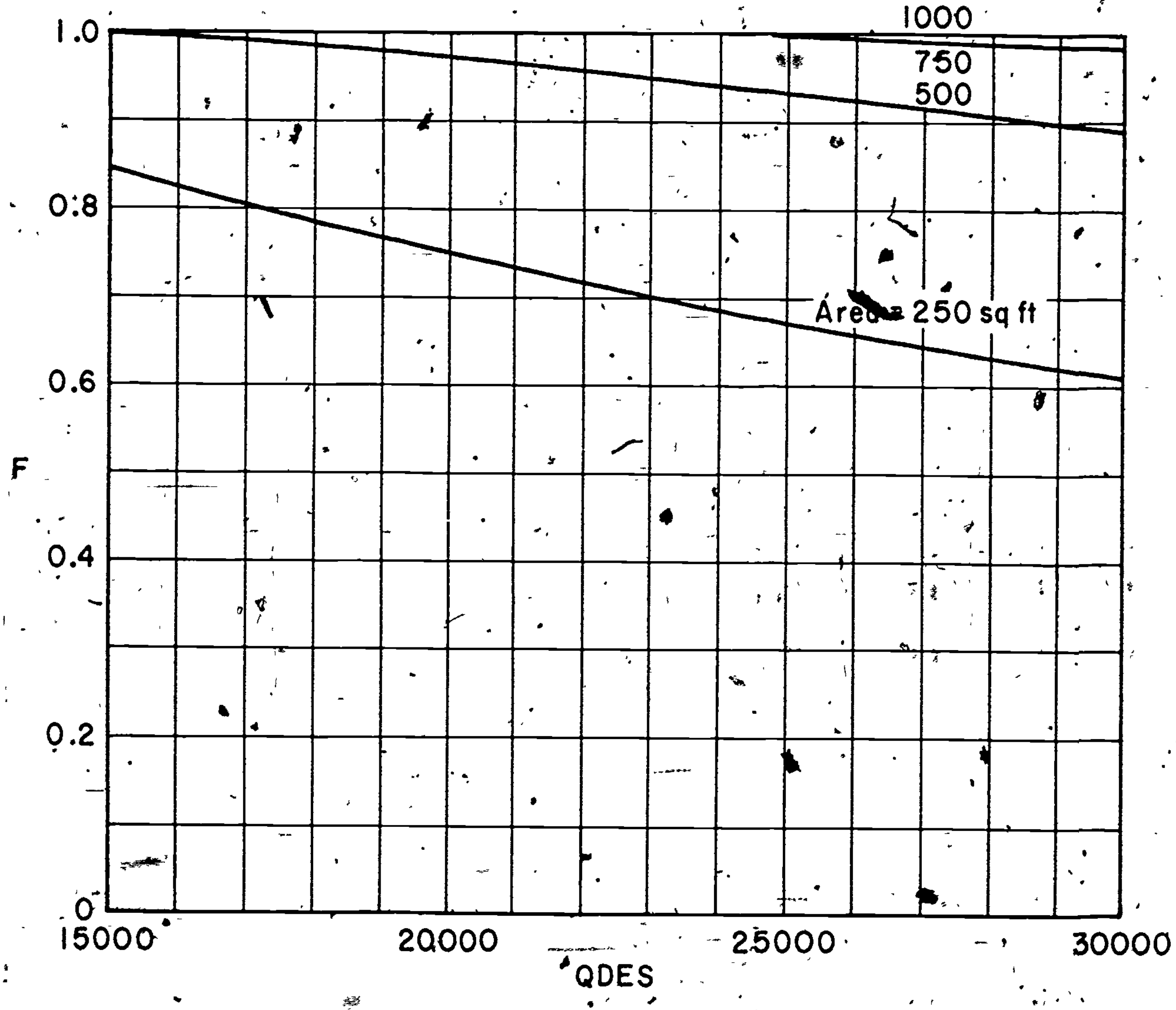

Figure 7-31.

AIR SYSTEM
SOLARON COLLECTOR
SLOPE = LAT + 15
LOCATION: SANTA MARIA

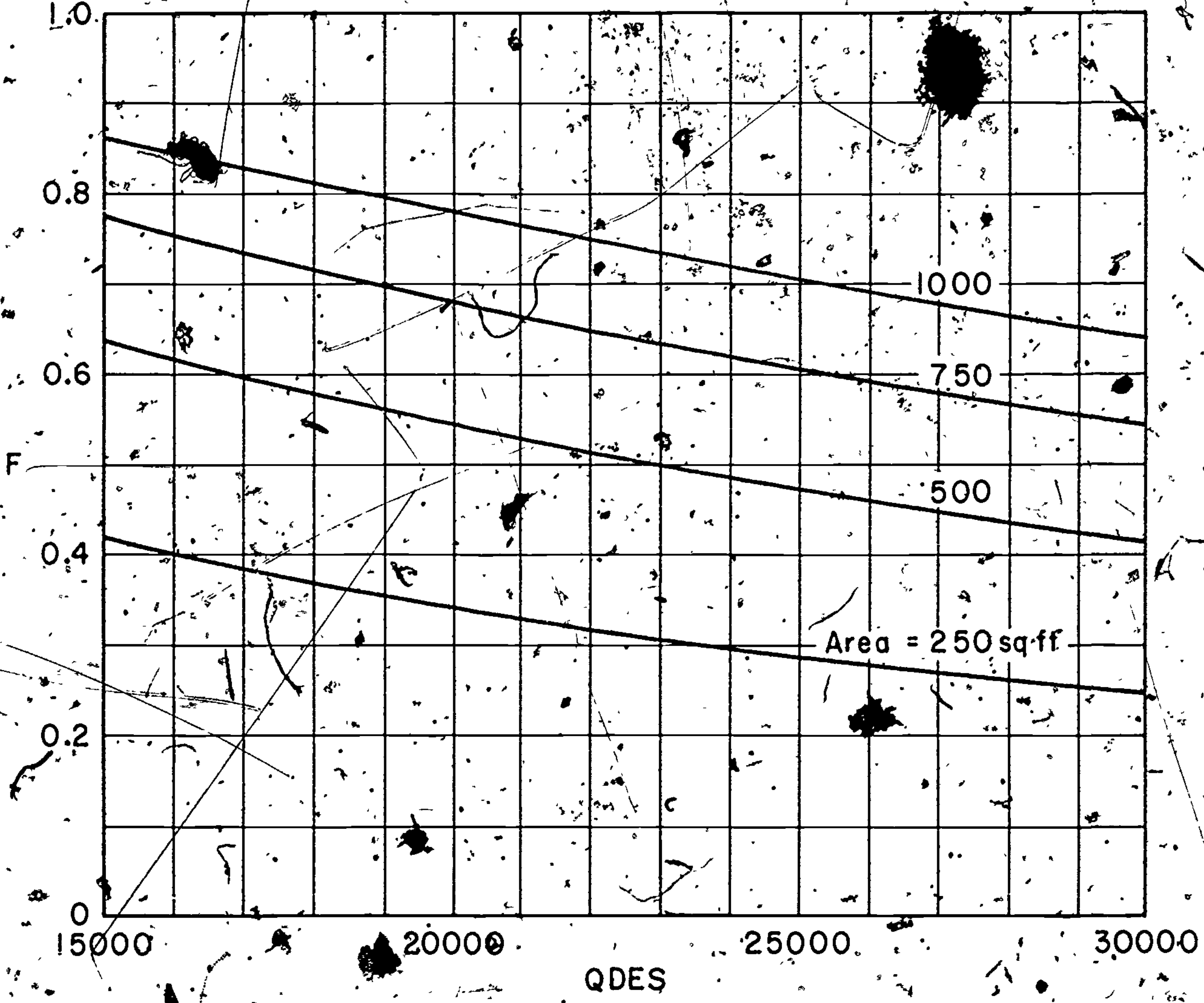

Figure 7-32.

AIR SYSTEM
SOLARON COLLECTOR
SLOPE = LAT + 15
LOCATION: WASH., D.C.

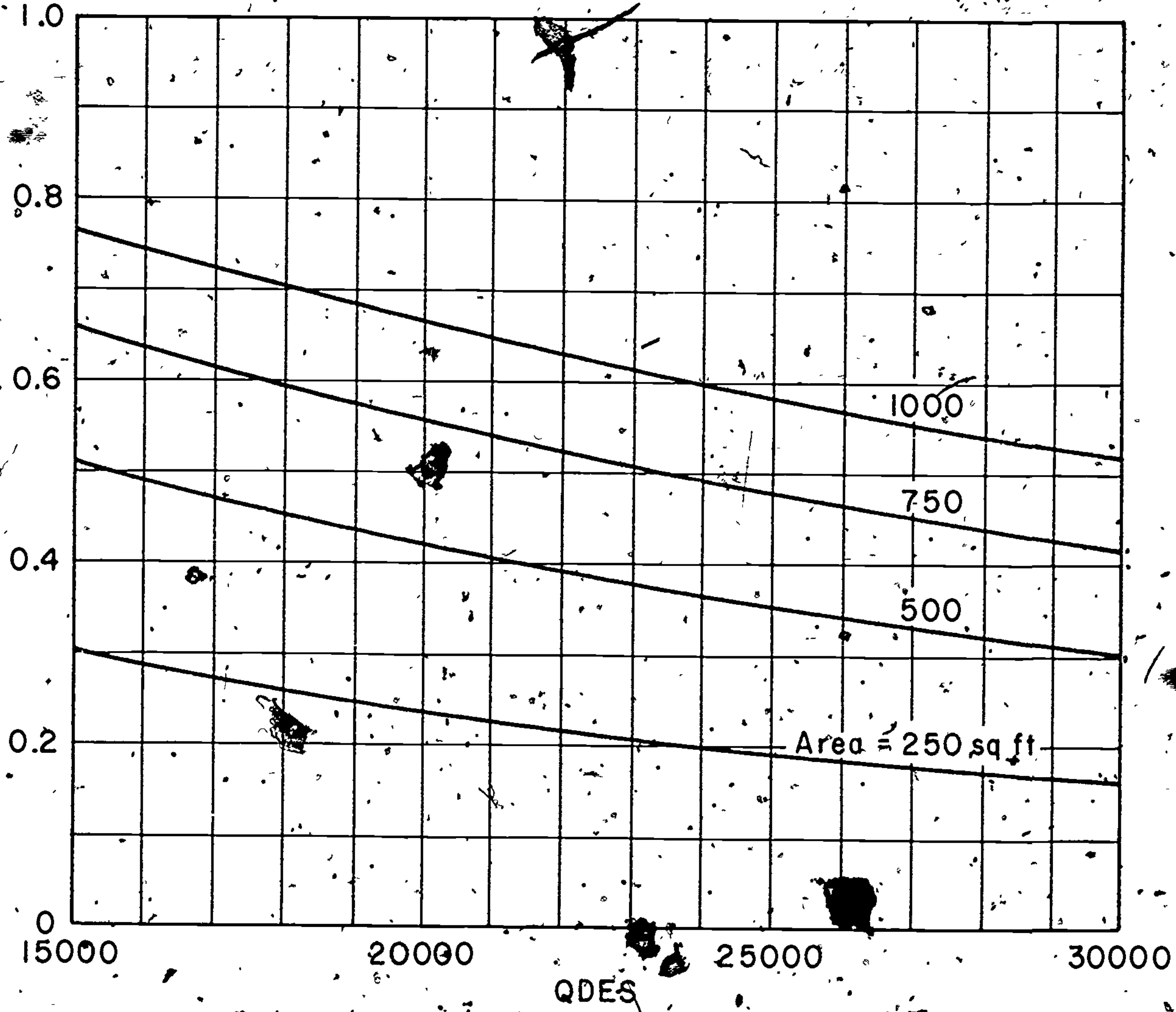

Figure 7-33

AIR SYSTEM
SOLARON COLLECTOR
SLOPE = LAT +15
ST. CLOUD

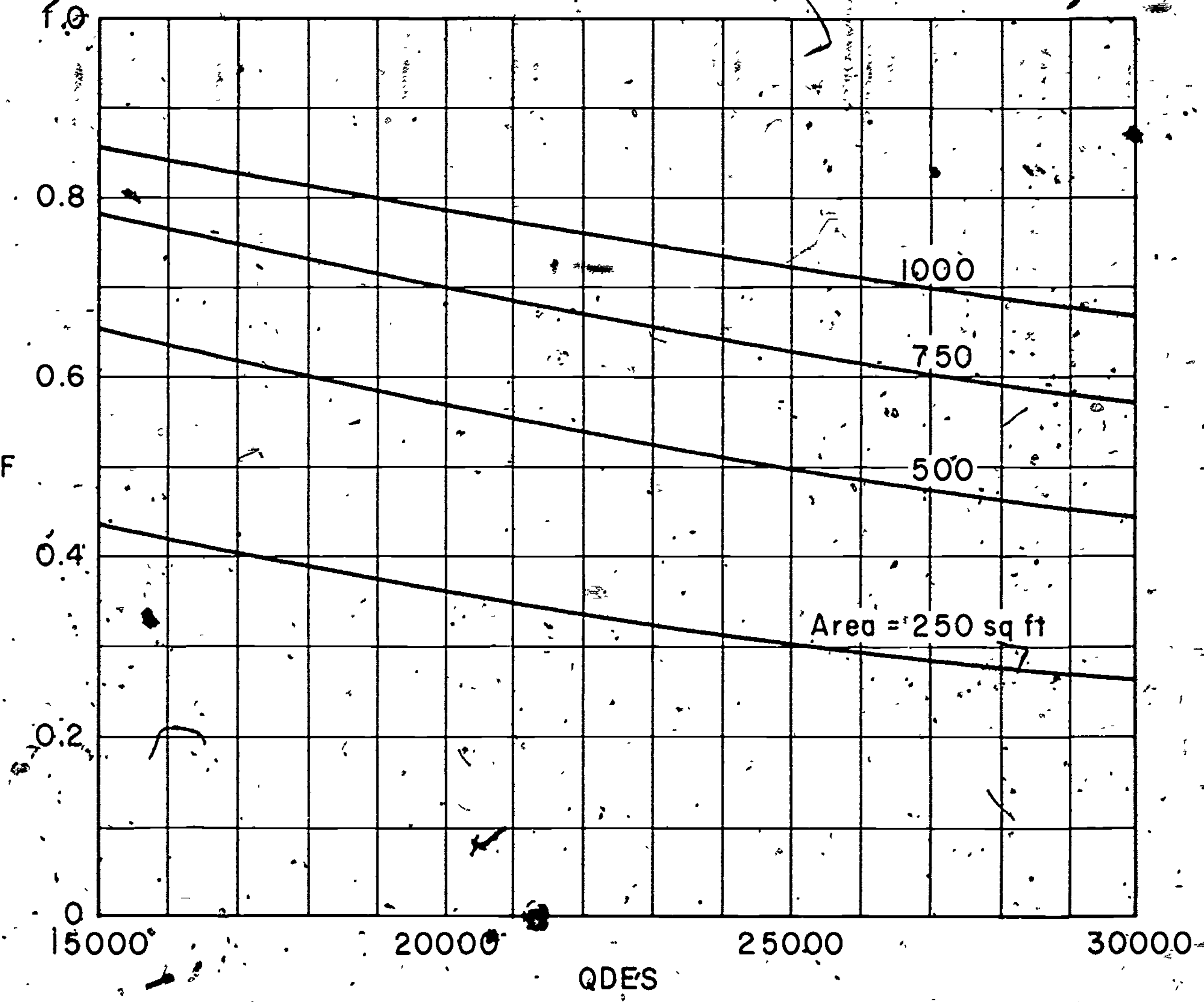

Figure 7-34.

WATER SYSTEM
PPG COLLECTOR
SLOPE = LAT
LOCATION: BOULDER

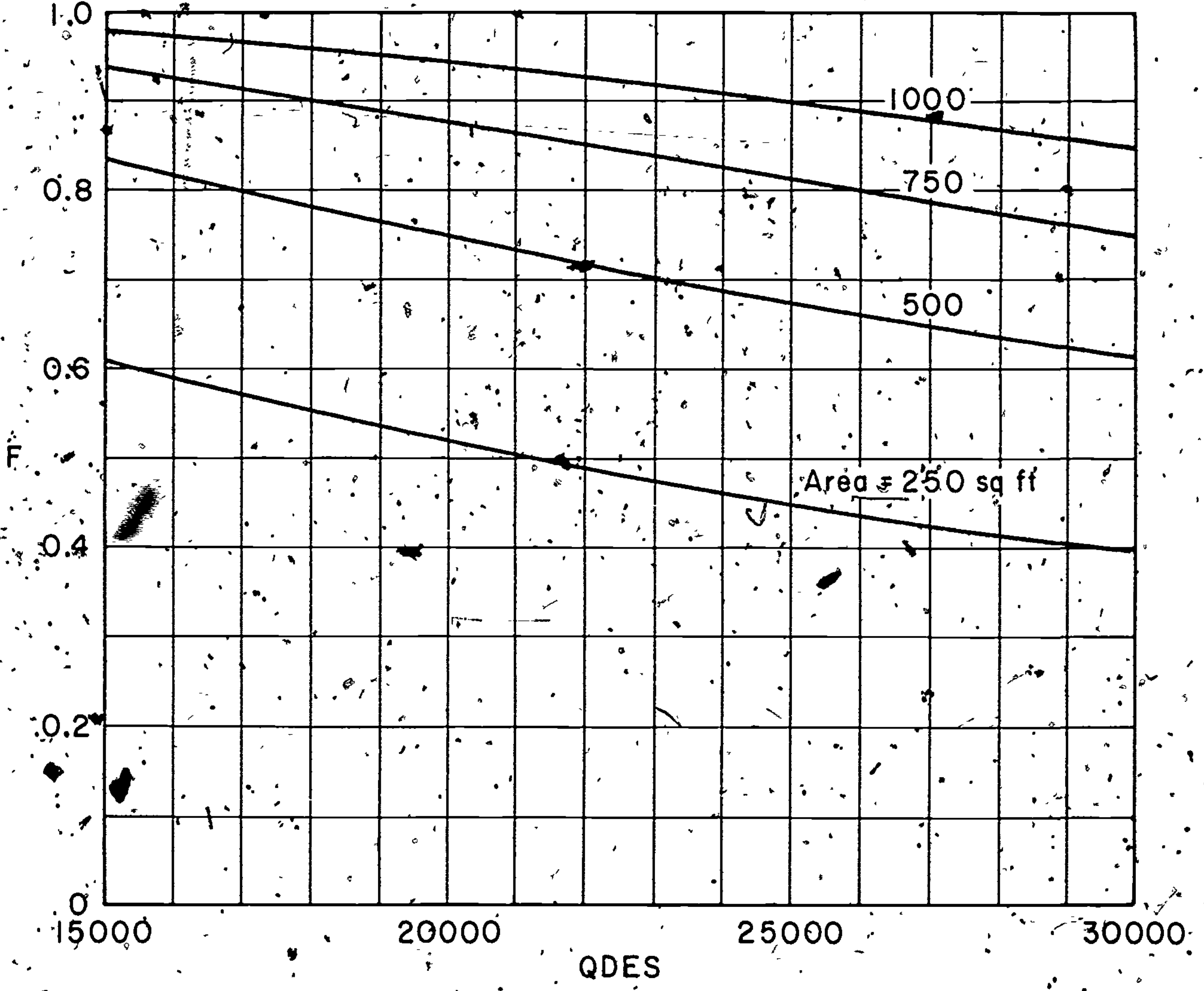

Figure 7-35.

WATER SYSTEM
PPG COLLECTOR
SLOPE = LAT
LOCATION: ALBUQUERQUE

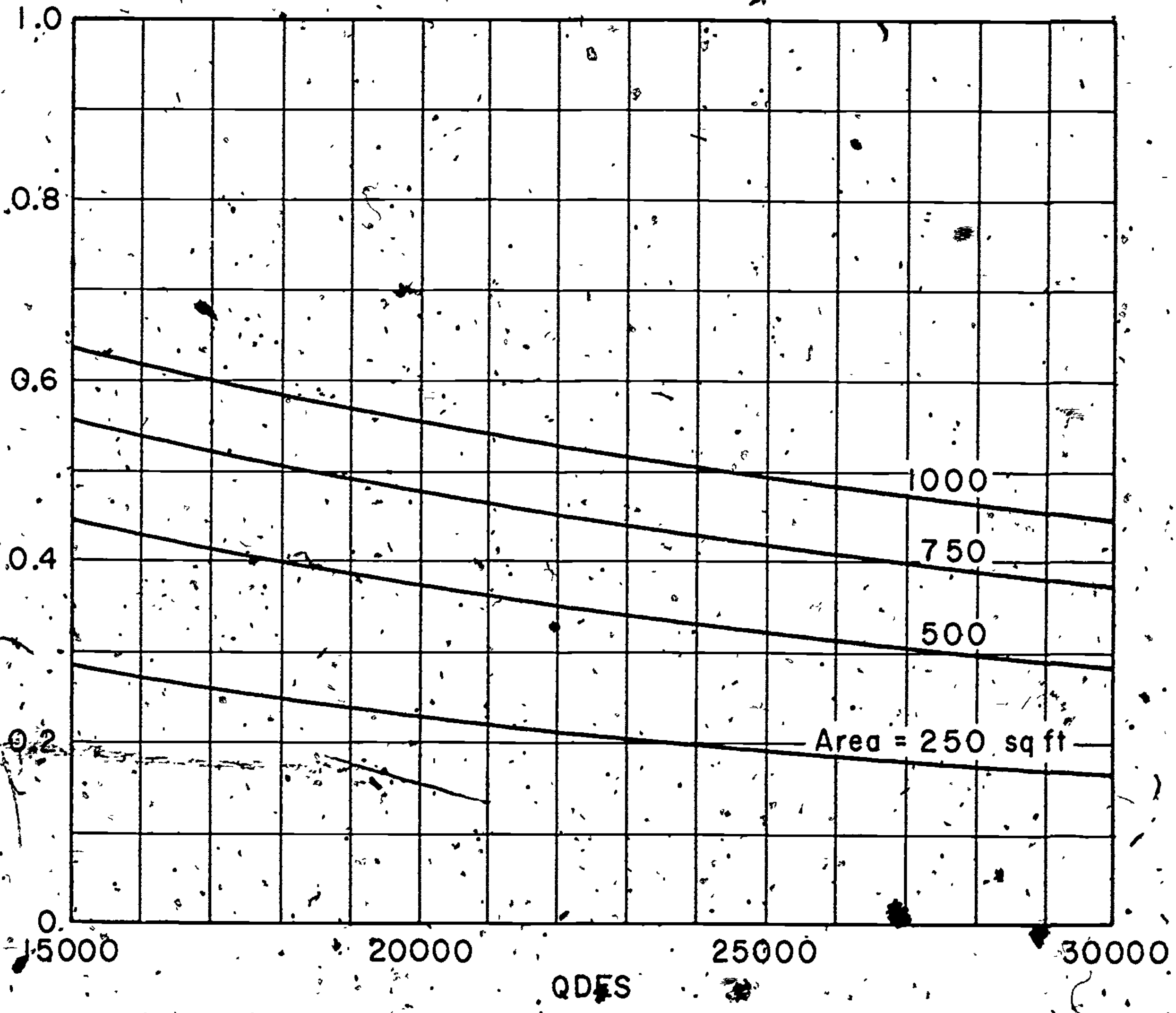

Figure 7-36.

WATER SYSTEM
PPG COLLECTOR
SLOPE = LAT
LOCATION: MADISON

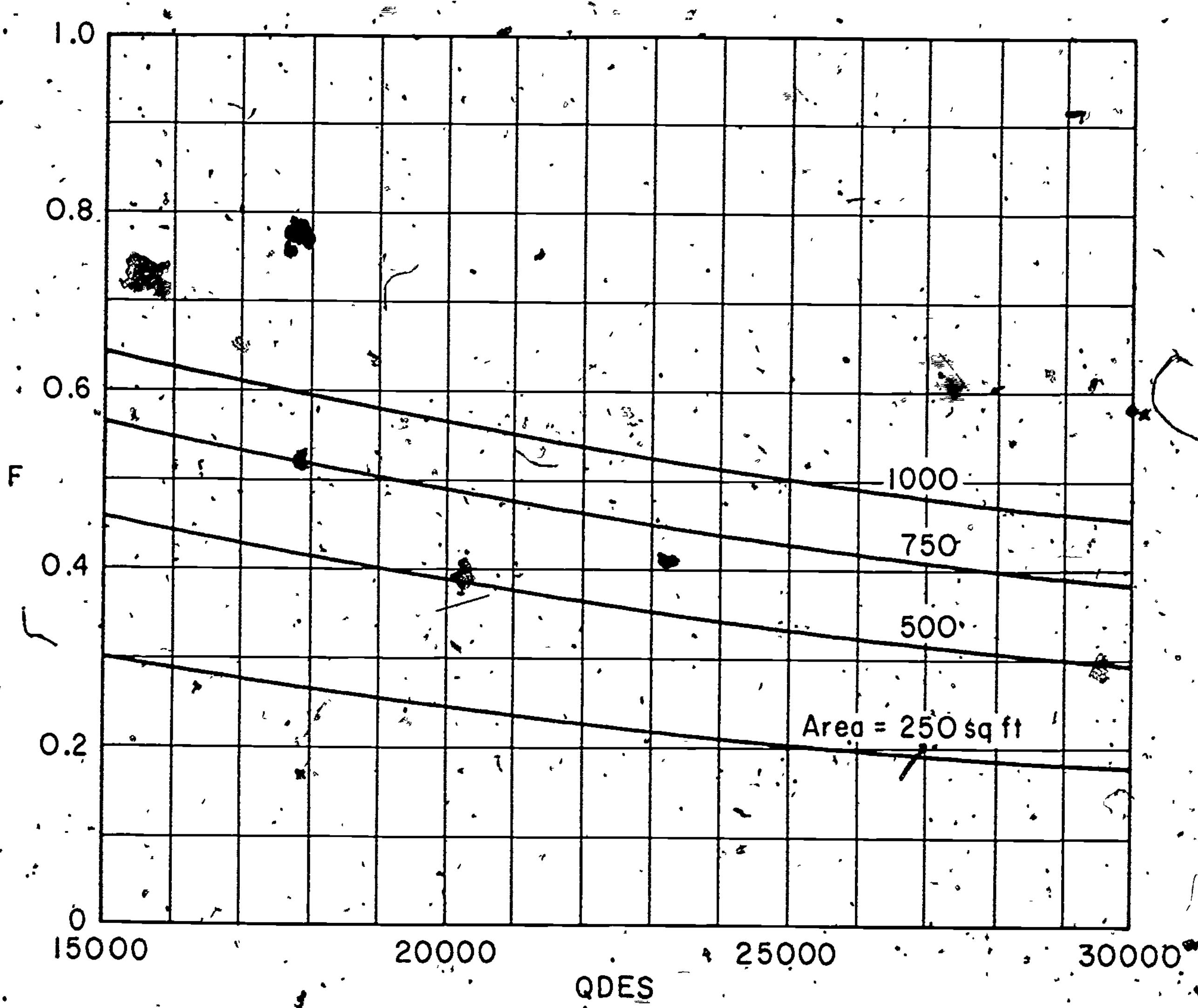

Figure 7-37.

WATER SYSTEM
PPG COLLECTOR
SLOPE = LAT
LOCATION: BOSTON

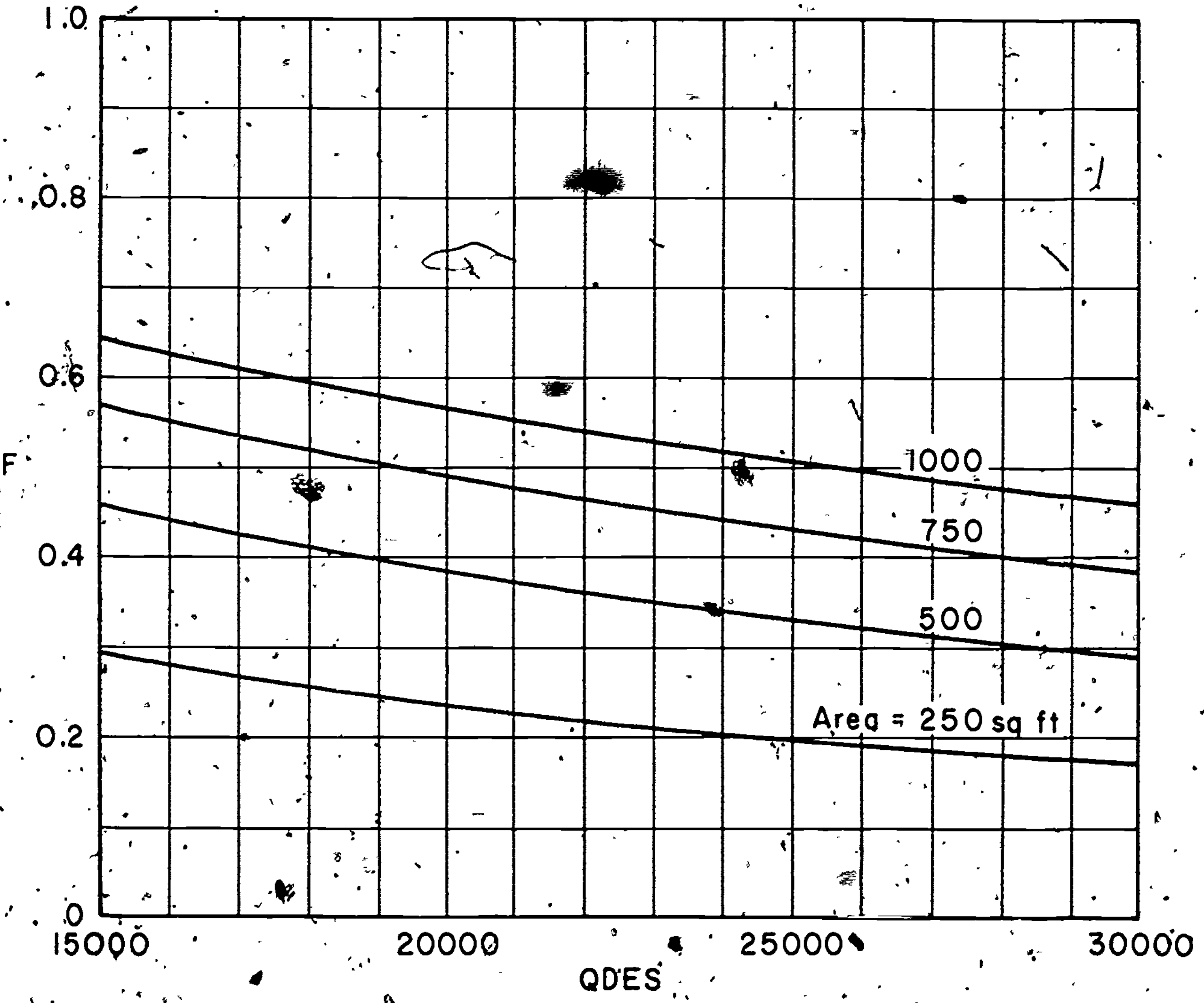

Figure 7-38.

WATER SYSTEM
PPG COLLECTOR
SLOPE = LAT
LOCATION: ALBANY

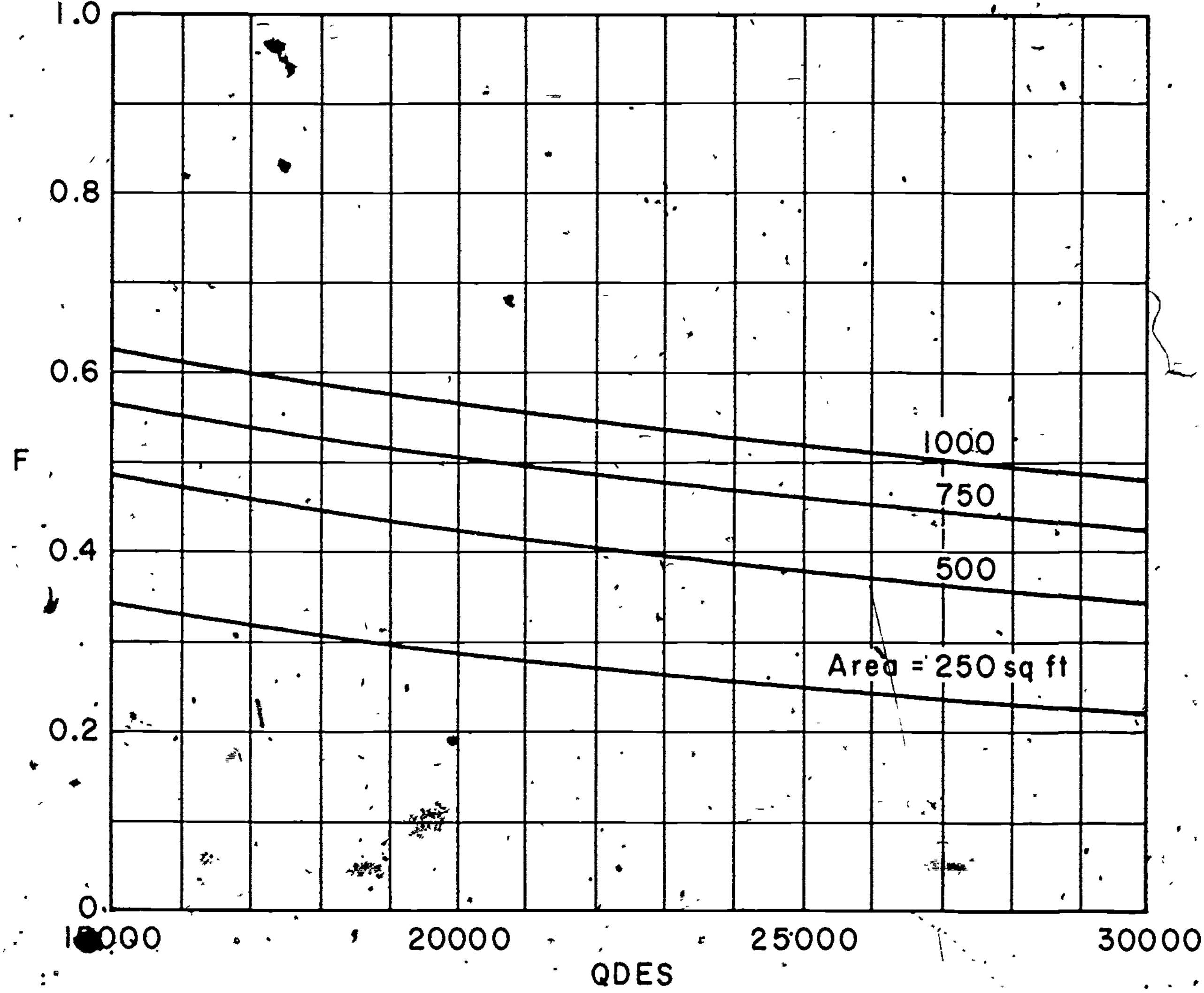

Figure 7-39.

WATER SYSTEM
PPG COLLECTOR
SLOPE = LAT
LOCATION: SEATTLE

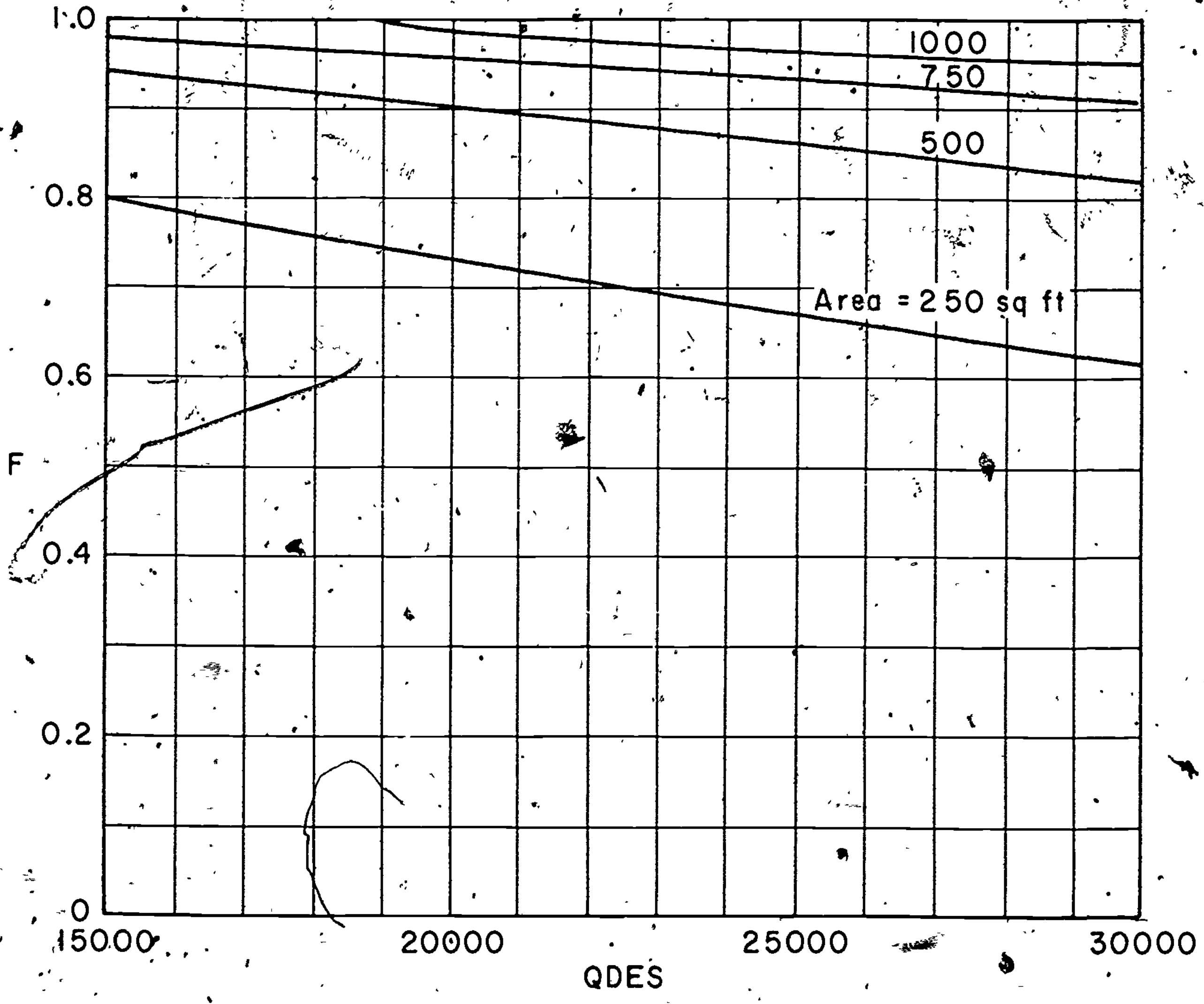

Figure 7-40.

WATER SYSTEM
PPG COLLECTOR
SLOPE = LAT
LOCATION: GAINESVILLE

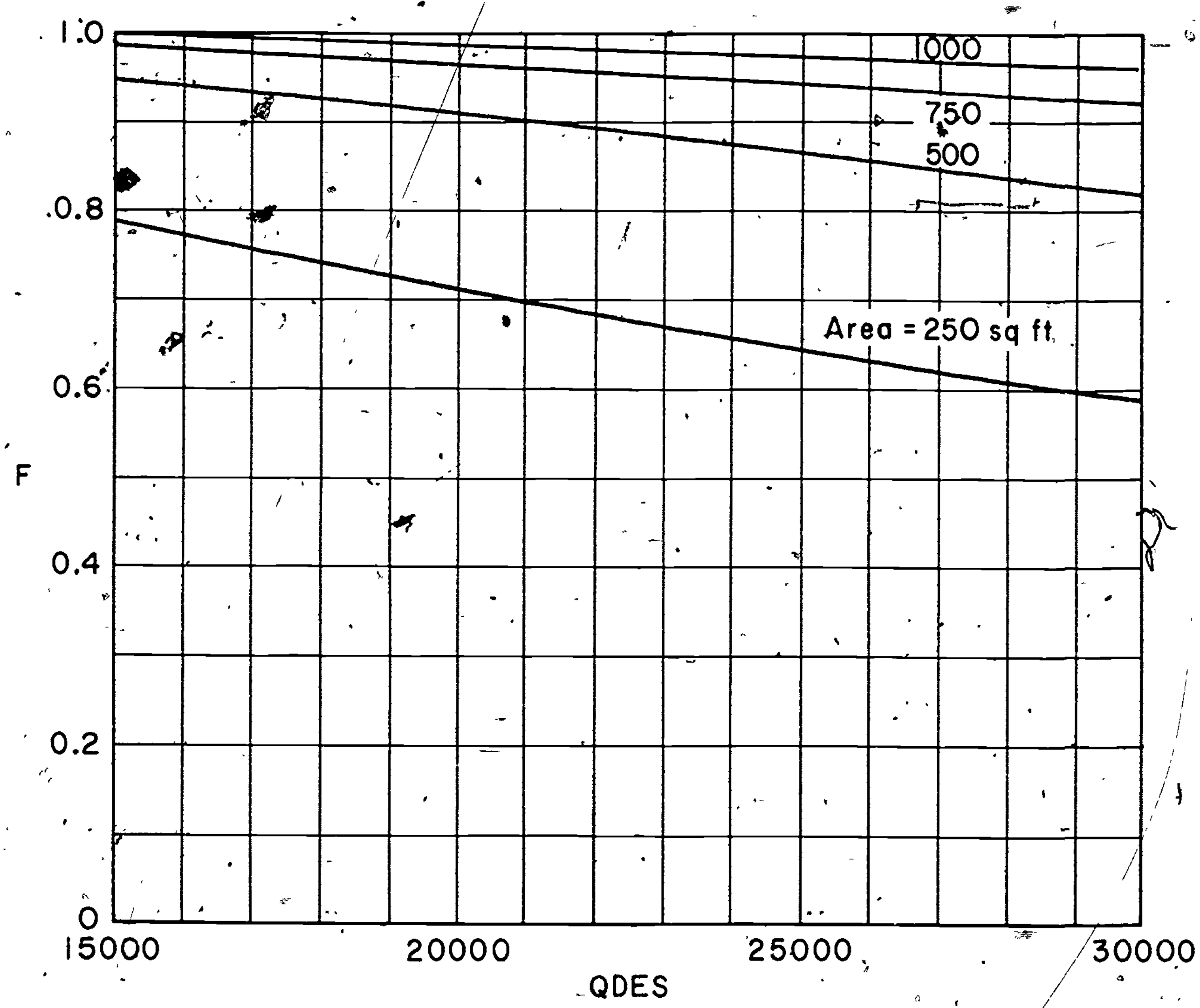

Figure 7-41.

WATER SYSTEM
PPG COLLECTOR
SLOPE = LAT
LOCATION: SANTA MARIA

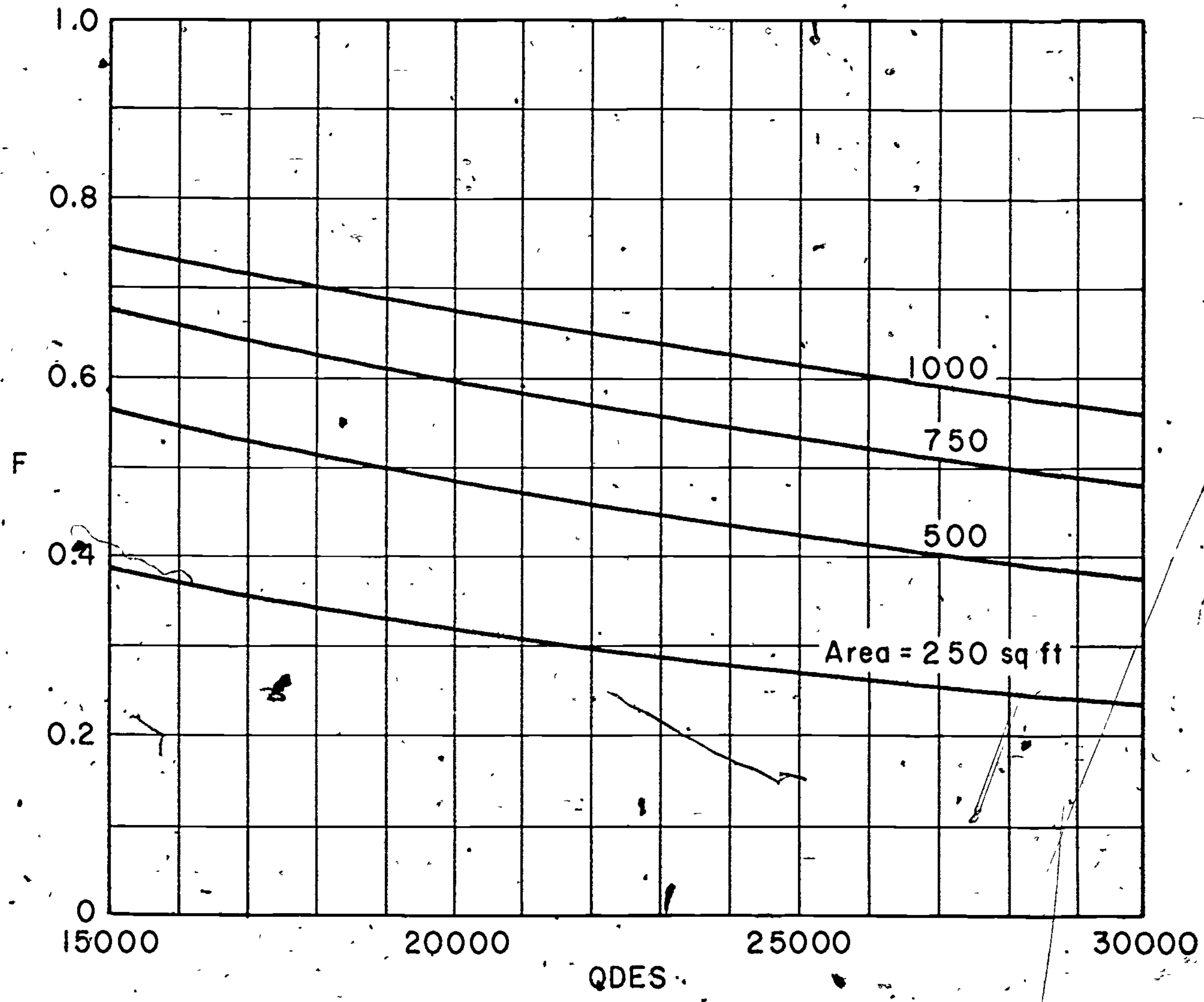

Figure 7-42.

WATER SYSTEM
PPG COLLECTOR
SLOPE = LAT
LOCATION: WASH., D.C.

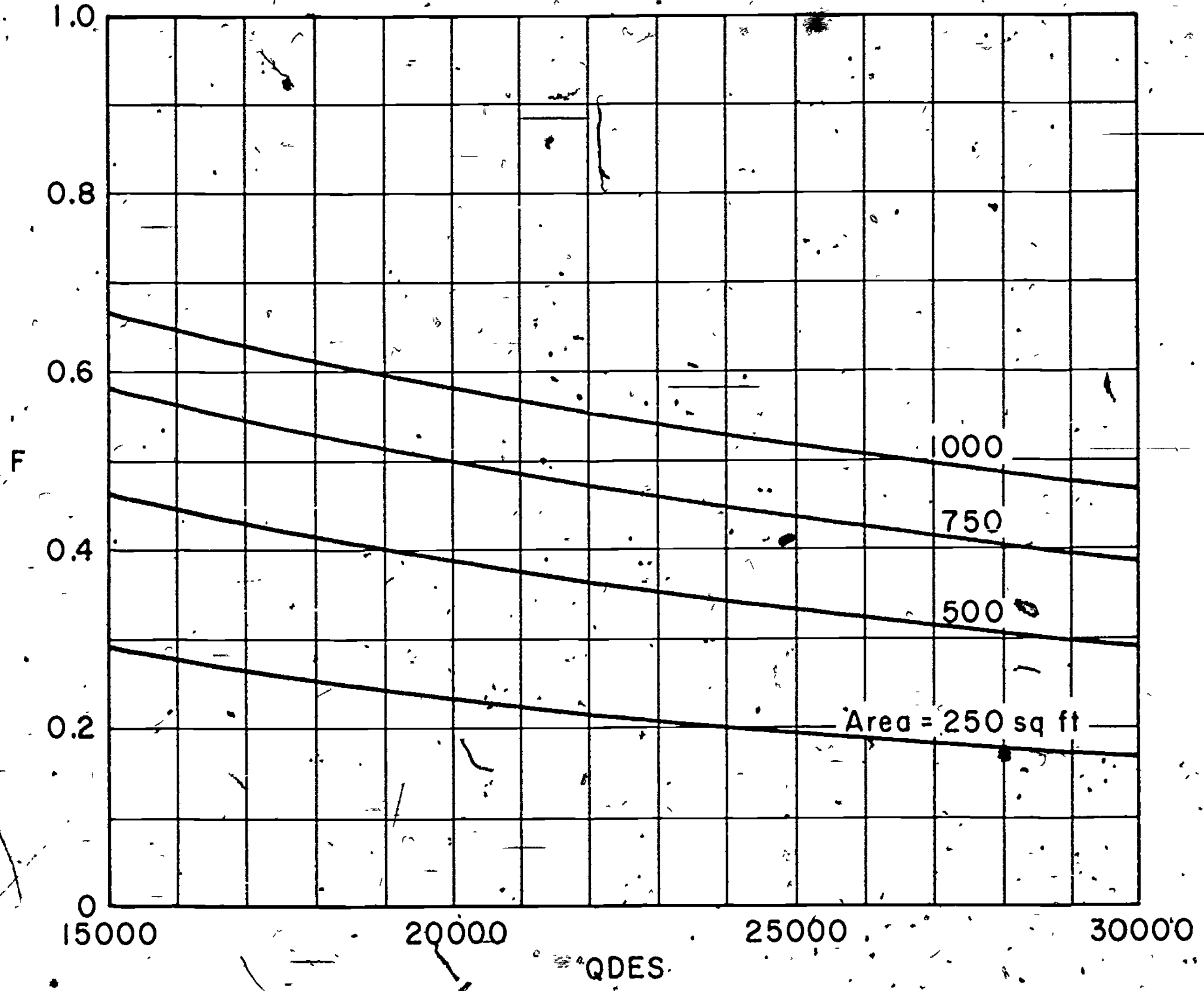

Figure 7-43.

WATER SYSTEM
PPG COLLECTOR
SLOPE = LAT
LOCATION: ST. CLOUD

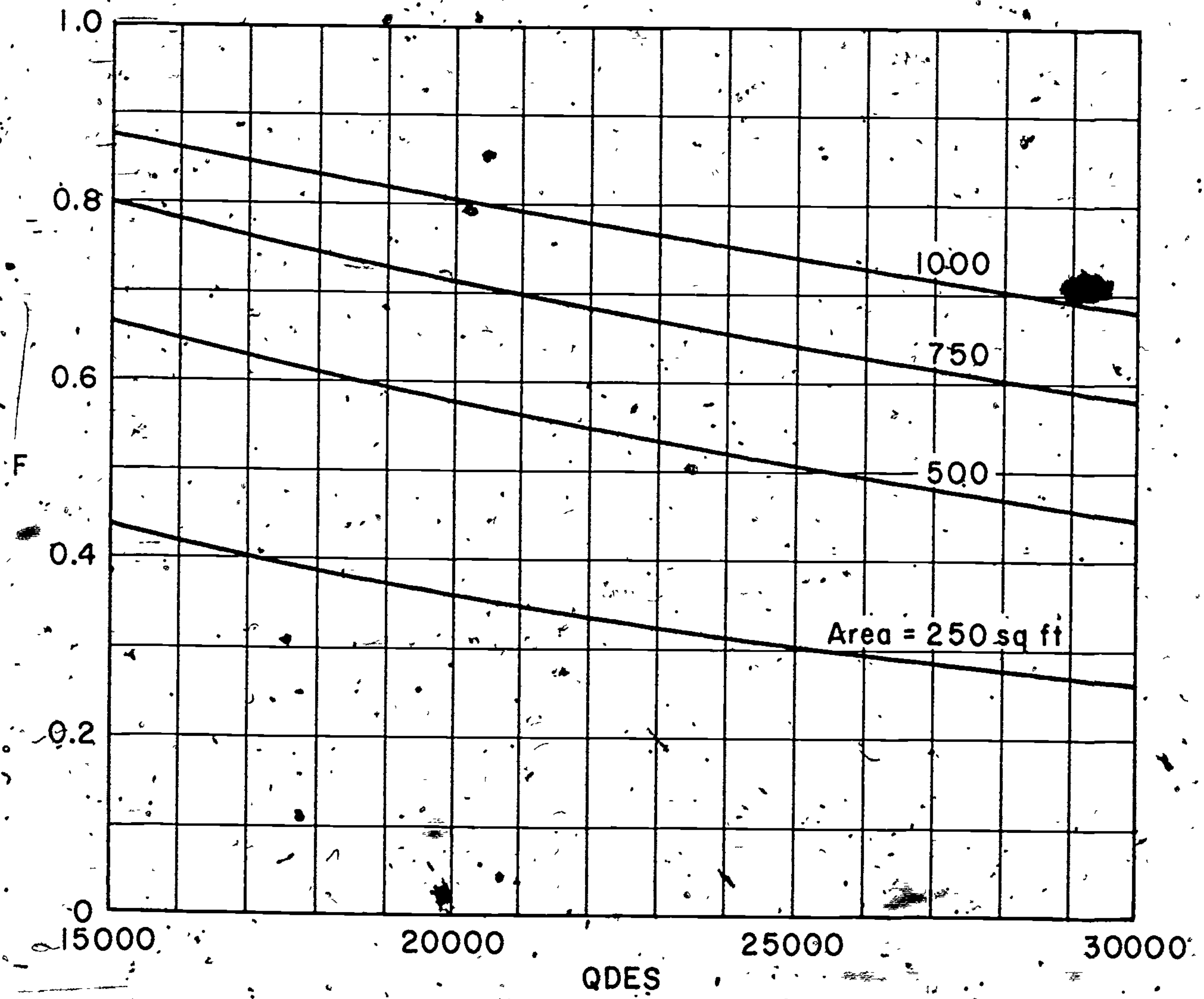

Figure 7-44.

WATER SYSTEM
PPG COLLECTOR
SLOPE = LAT + 15
LOCATION: BOULDER

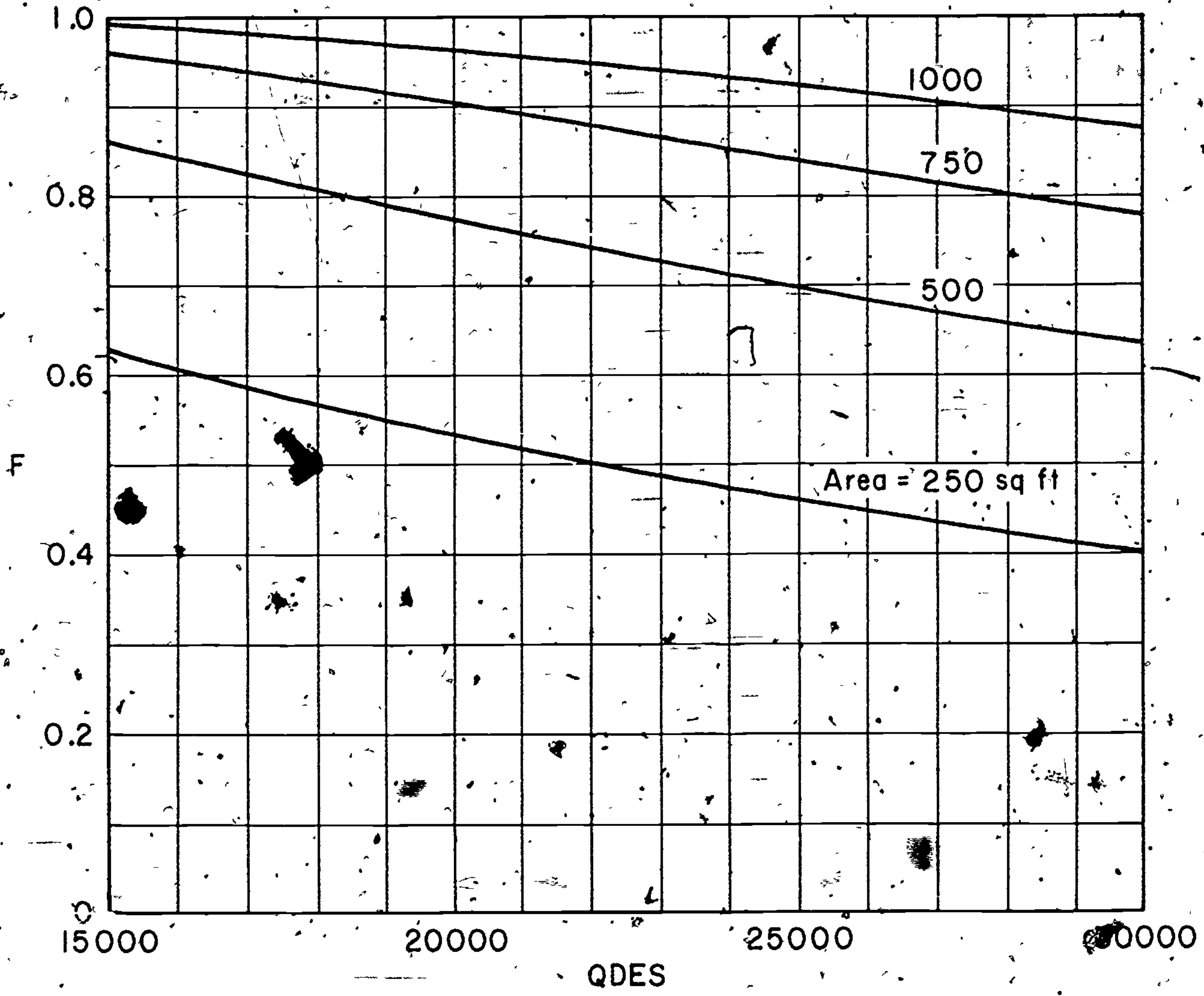

Figure 7-45.

WATER SYSTEM
PPG COLLECTOR
SLOPE = LAT + 15
LOCATION: ALBUQUERQUE

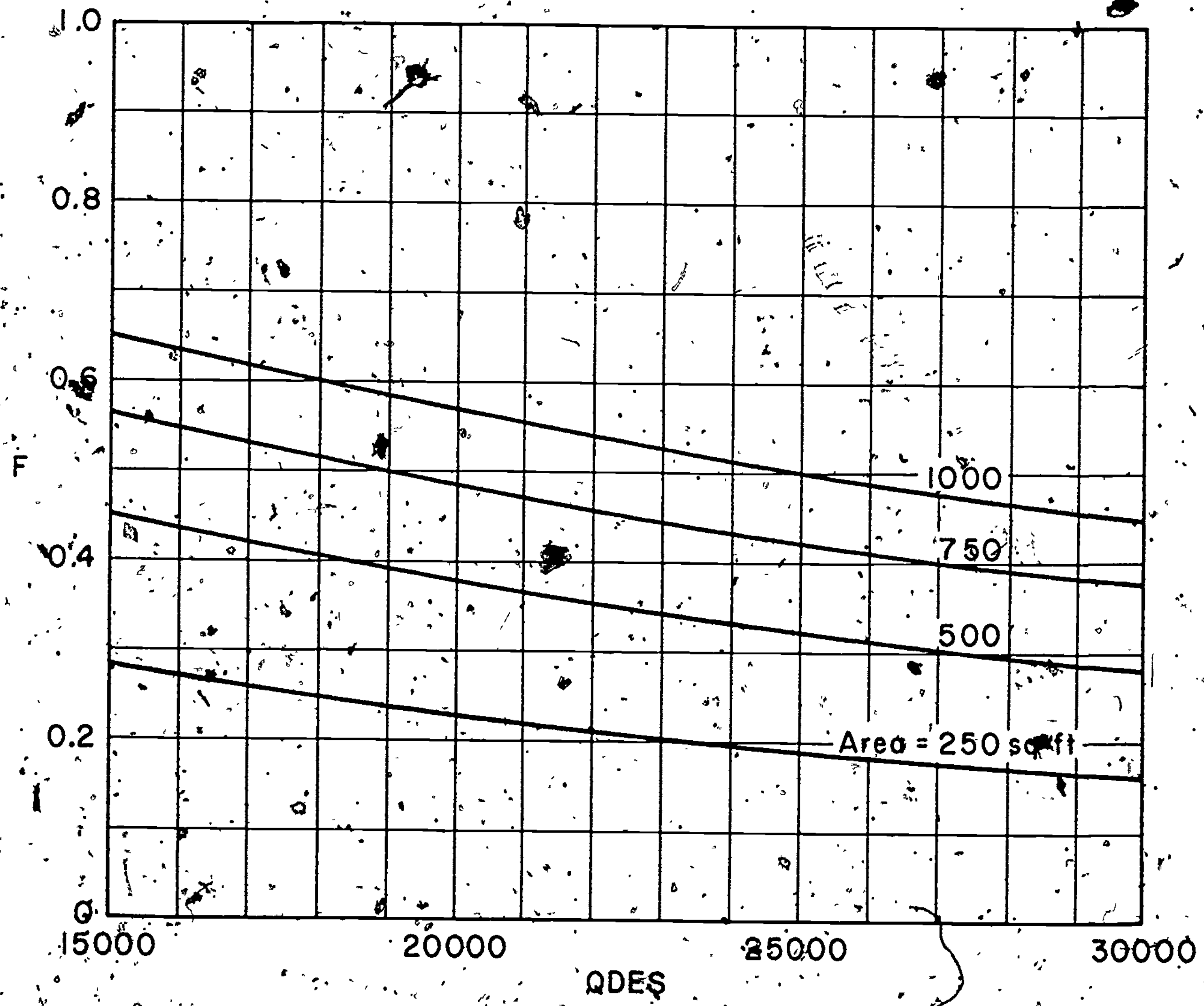

Figure 7-46.

WATER SYSTEM
PPG COLLECTOR
SLOPE = LAT + 15
LOCATION: MADISON

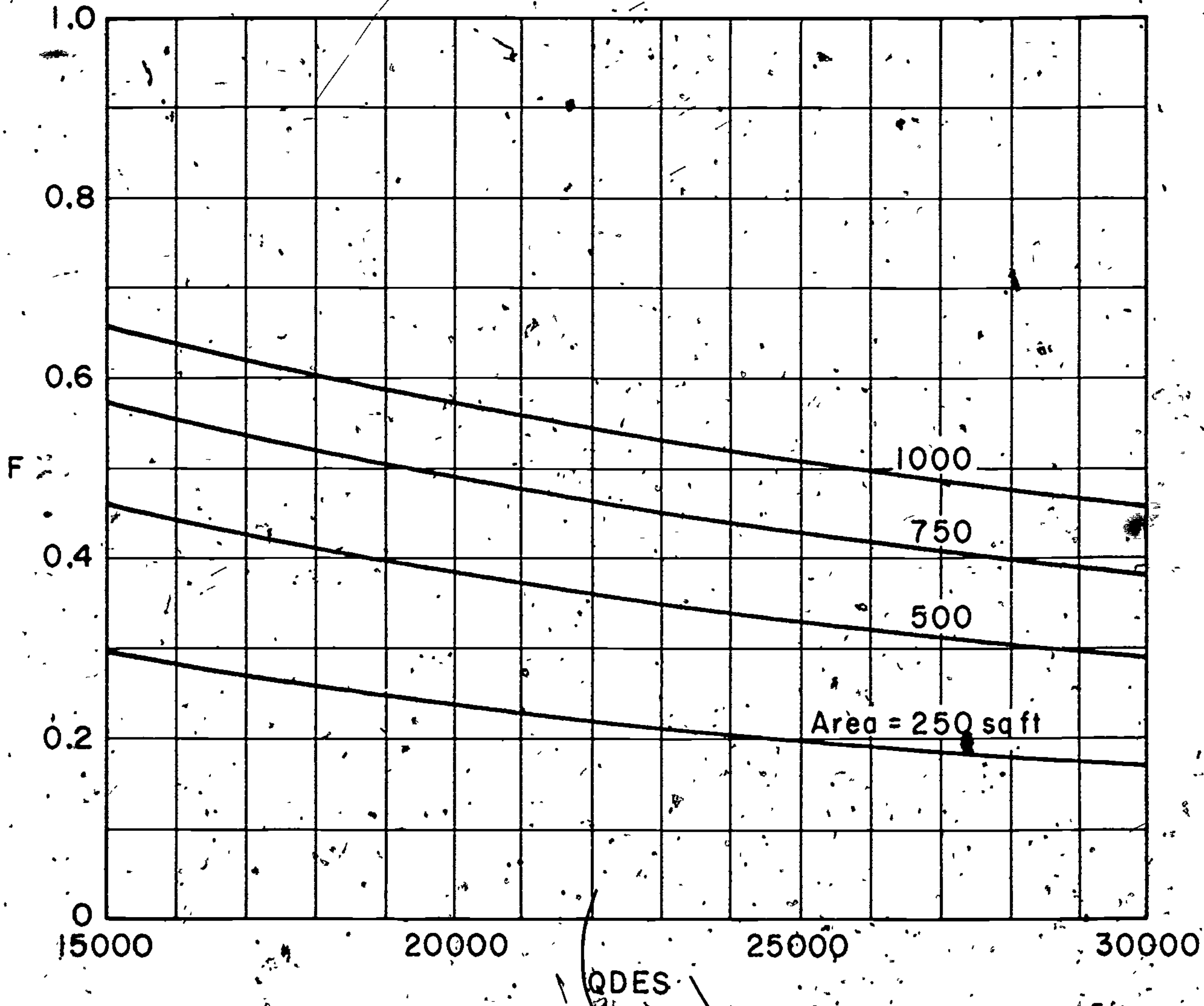

Figure 7-47.

WATER SYSTEM
PPG COLLECTOR
SLOPE = LAT + 15
LOCATION: BOSTON

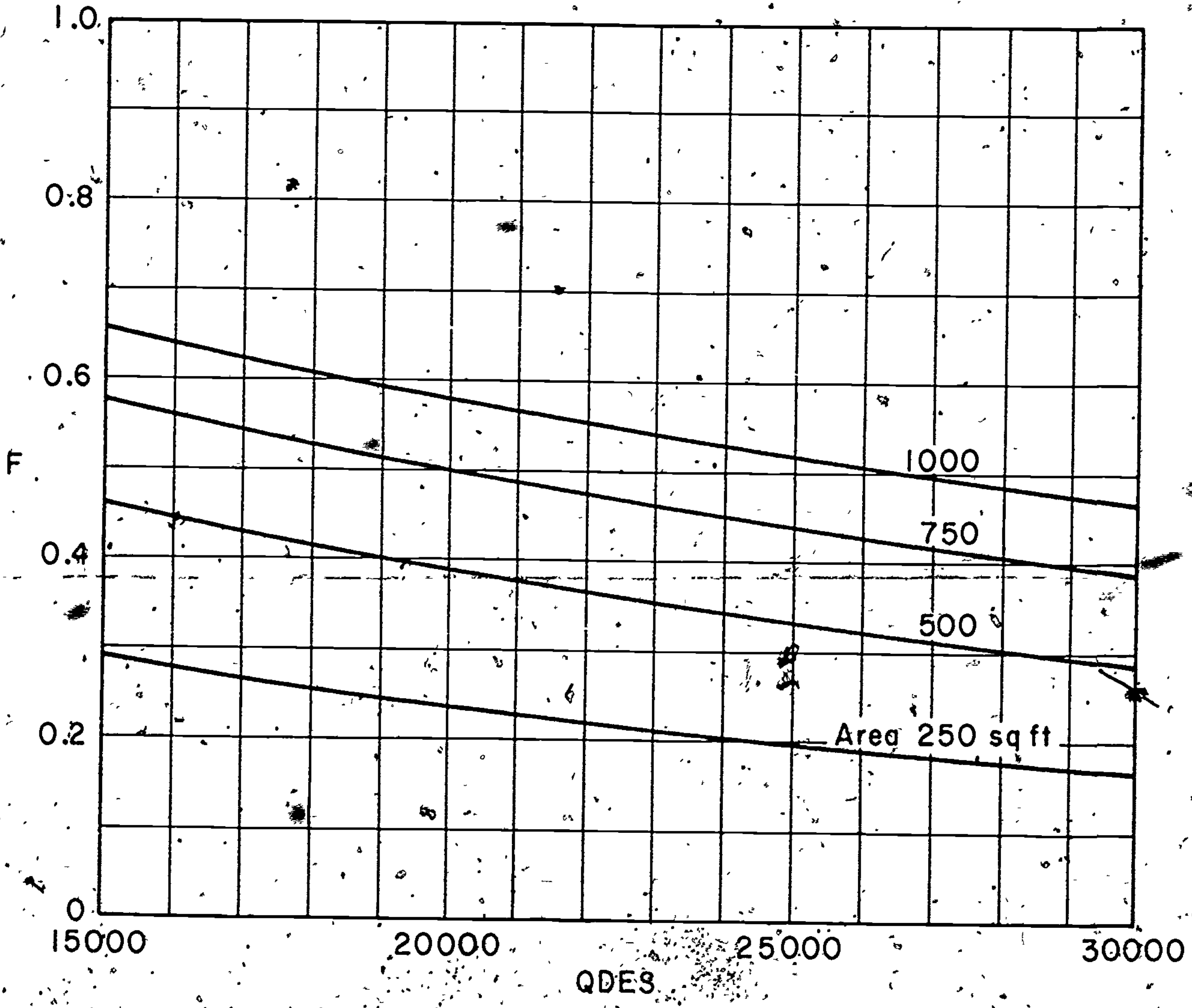

Figure 7-48.

WATER SYSTEM
PPG COLLECTOR
SLOPE = LAT + 15
LOCATION: ALBANY

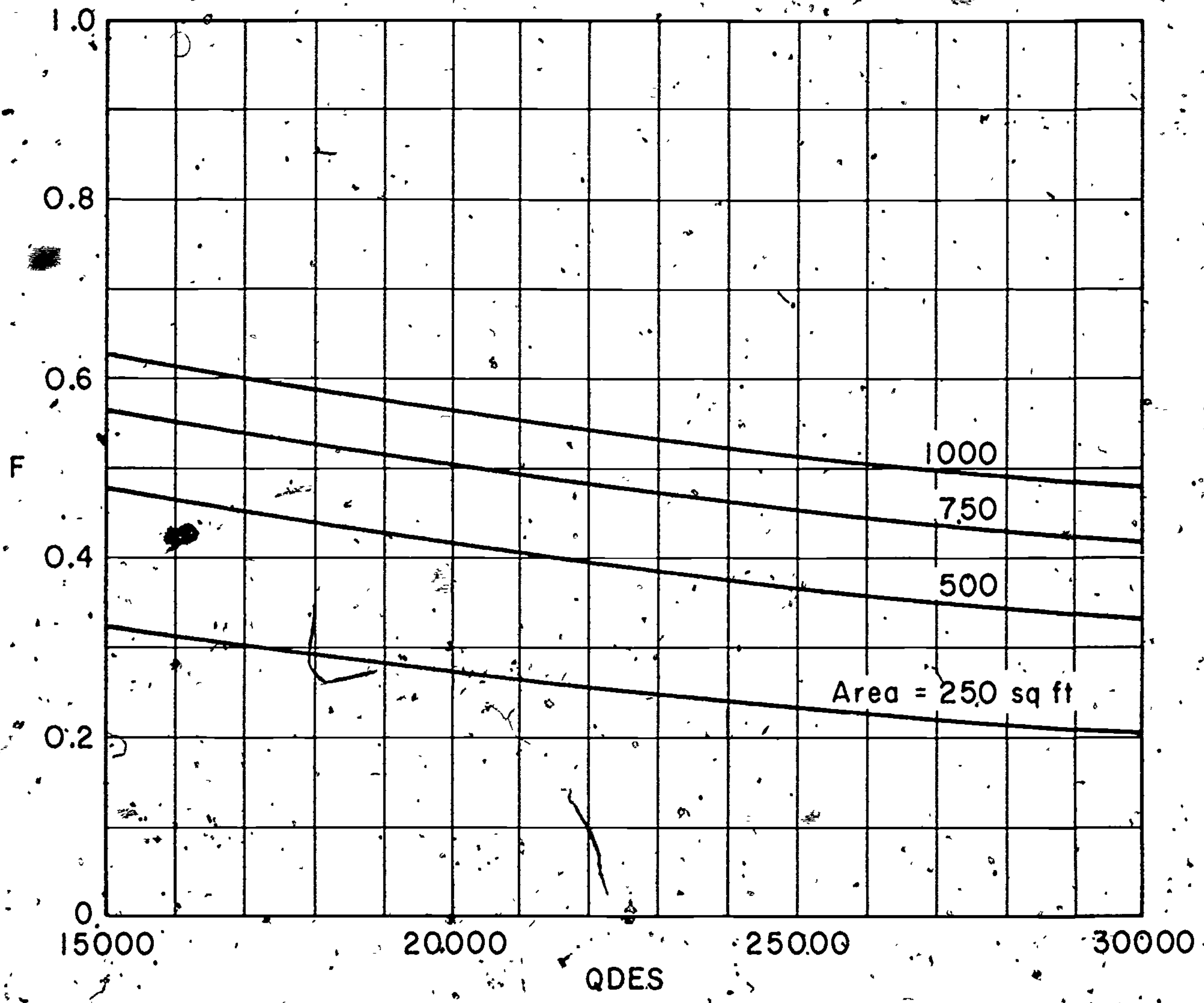

Figure 7-49.

WATER SYSTEM
PRG COLLECTOR
SLOPE = LAT + 15
LOCATION: SEATTLE

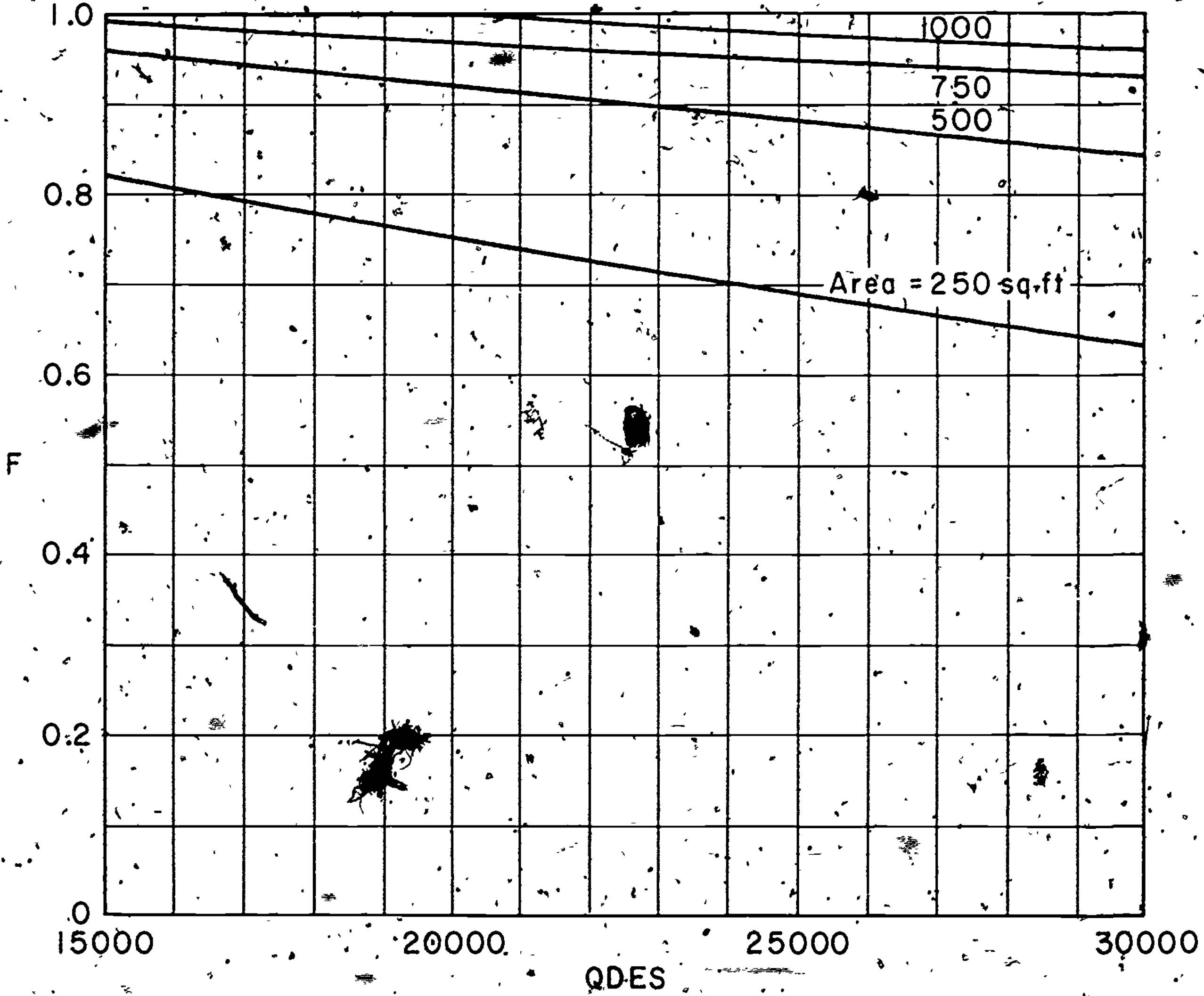

Figure 7-50.

WATER SYSTEM
PPG COLLECTOR
SLOPE = LAT. + 15
LOCATION: GAINESVILLE

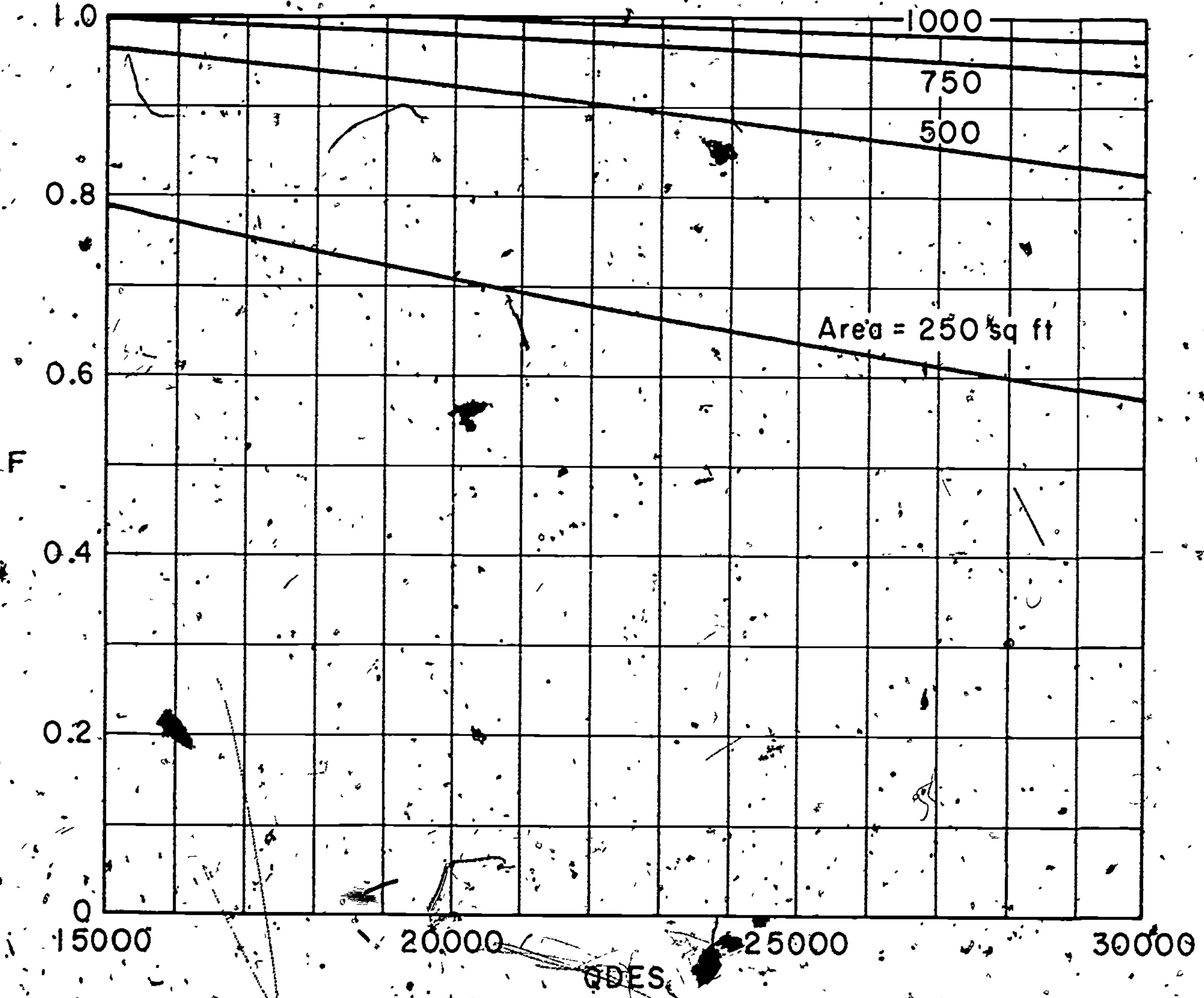

Figure 7-51.

WATER SYSTEM
PPG COLLECTOR
SLOPE = LAT + 15
LOCATION: SANTA MARIA

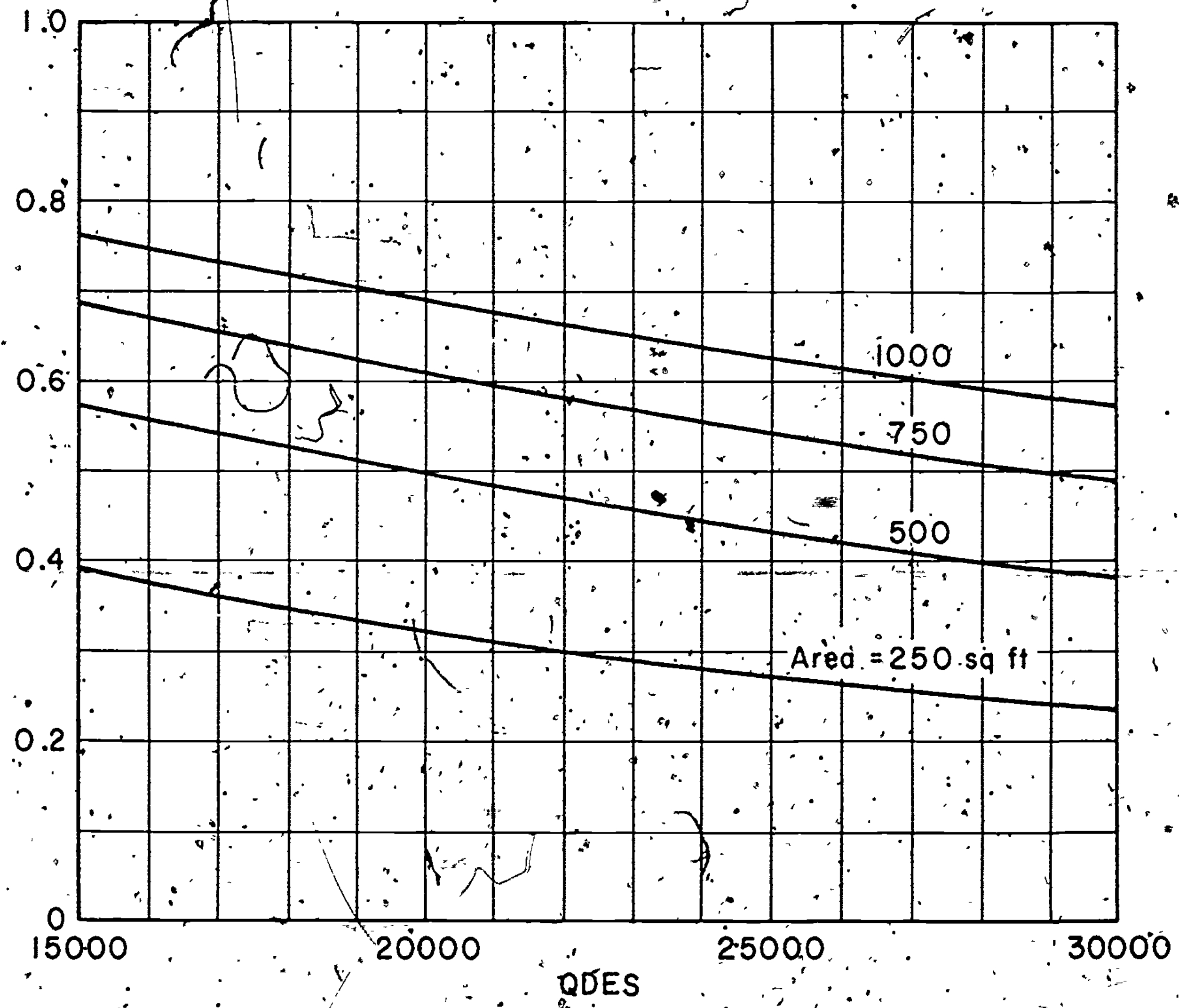

Figure 7-52.

WATER SYSTEM
PPG COLLECTOR
SLOPE = LAT + 15
LOCATION: WASH., D.C.

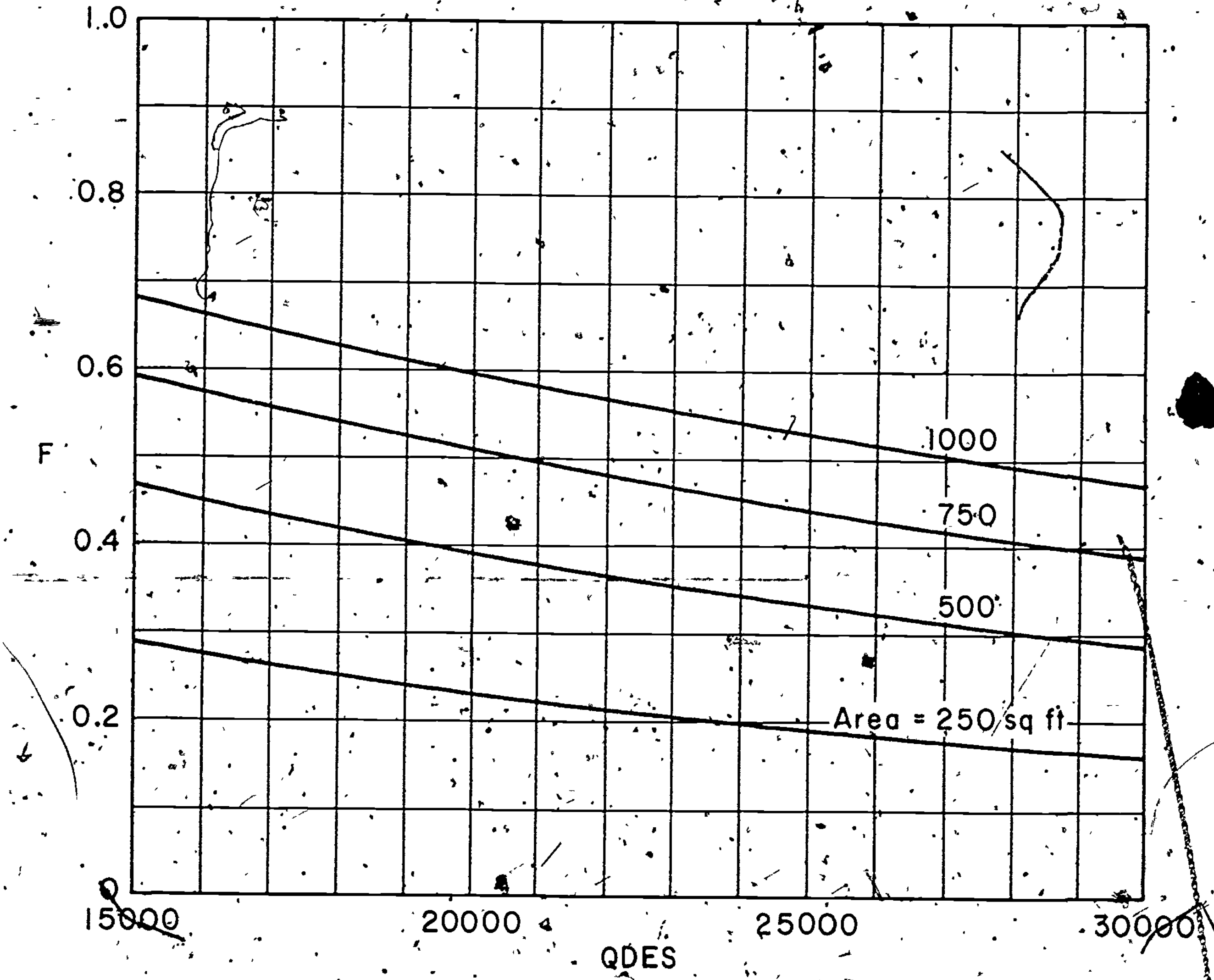

figure 7-53.

WATER SYSTEM
PPG COLLECTOR
SLOPE = LAT + 15
LOCATION; ST. CLOUD

TRAINING COURSE IN
THE PRACTICAL ASPECTS OF
DESIGN OF SOLAR HEATING AND COOLING SYSTEMS
FOR
RESIDENTIAL BUILDINGS

MODULE 8

ECONOMIC CONSIDERATIONS

SOLAR ENERGY APPLICATIONS LABORATORY
COLORADO STATE UNIVERSITY
FORT COLLINS, COLORADO

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LIST OF SYMBOLS

i	interest rate on money, decimal
n	mortgage term, years
P	present worth, dollars
f	percent of annual heating requirement provided by Solar
F	future value, dollars

INTRODUCTION

This module presents a methodology that may be used to determine the economics of solar heating systems. It also presents current cost figures for some solar systems utilizing both water and air.

There are many different approaches that have been taken to determine economics of solar systems. Many of these have been used to illustrate favorable economics of solar systems and are often unrealistic. The methodology presented in this module may be applied to any system and the user can provide relevant cost information in his locality.

TRAINEE-ORIENTED OBJECTIVE

The objective in the module is to present current system costs and a methodology that will enable the trainee to determine the economies of solar systems.

SUB-OBJECTIVES

From the costs and methodology presented, the trainee will be able to:

1. Estimate system costs
2. Compare solar system costs with conventional system costs
3. Explain economic calculations to his clients.

SYSTEM COSTS

Examination of the costs of a number of solar systems that have been constructed within the last three years indicates installed system costs to be approximately $25 per square foot of collector. This is based upon systems that have been designed to carry approximately 75 percent of the total heating and service hot water loads for single family residential type structures. This unit-installed cost applies to both air and water systems. It is a current cost (1976) based on Colorado prices, and for detailed economic calculations in other areas it is cautioned that local estimates of system costs should be made.

Tables 8-1 and 8-2 present actual cost data for a water system and an air system that have been recently designed and installed. Cost items for drawings and design time are included in the profit and overhead figure. Such costs are real, however, and must be included in any economic analysis. It should be noted that the collector cost shown in Table 8-1 is below the average cost for collectors, and the control cost exceeds the average cost for controllers. These tables do not include freight charges, as these charges would vary with respect to location. They can easily represent a significant addition to the total costs, however. The size of the collector array is different for the water system shown in Table 8-1 than for the air system shown in Table 8-2 because of the different design heat loads. The ratio between design heat load and collector size is comparable for the two cases.

We can see from the two cases illustrated that the installed cost of an air system is greater than for the water system. However, some adjustment in the unit cost of the smaller air system should be made, and it is reasonable to expect that for equivalent sizes the costs would be between $20 and $25 dollars per square foot of collector.

The homeowner is most likely to think of solar energy as a means of reducing future heating costs. What he would like to do is add the initial cost of solar components to his mortgage and make the additional mortgage payments from the future fuel savings.

Table 8-1. Representative Costs for a Water System with 800 ft^2 Collector

House Design Heat Load = 110,000 Btu/hr

ITEM		COST	
1.	Collectors	$4,080	
2.	1200 gallon steel storage tank (lined)	1,400	
3.	Pumps - 350 80 185 130	745	
4.	Heat exchangers - 275 185	460	
			$ 6,685
5.	Plumbing - 1-1/4" pipe	230	
	Misc. fittings	220	
	Valves	536	
	Flow regulators	160	
	Manifolds	180	
	Cu pump	100	
	Air vents	20	
			1,446
6.	Controls and display	1,148	
7.	Insulation	1,481	
8.	Preheat tank	160	
9.	Expansion tank	80	
10.	Labor	3,600	
11.	Testing, balancing, adjusting, and periodic checks	1,000	
	Subtotal		15,600
12.	Profit and overhead (20% of $15,600)	3,120	
	TOTAL COST		$18,720

Installed cost per unit area of collector $23.40

Table 8-2. Representative Costs for an Air System with 390 ft^2 of Collector

House Design Heat Load = 42,000 Btu/hr

ITEM	COST
1. Solar equipment collectors, cap strips, end caps, air handler, controller, domestic hot water package, butyl sealant	$ 6,315.69
2. Engineering charge	400.00
3. Storage	
materials	500.00
labor	500.00
4. Gas hookup	80.00
5. Installation collectors, ductwork, controls, air handling unit, etc.	3,101.54
TOTAL COST	$10,897.23

Installed cost per unit area of collector $27.94

The economics of solar heating will be evaluated for an actual system under construction in the vicinity of Fort Collins, Colorado, to determine if the prospective fuel savings would be sufficient to repay the added mortgage costs. The cost of the solar system was $12,000. An annual payment to repay $12,000 over 25 years at 9-percent interest (the interest on the loan) is $1221.68, which would be $101.80 per month. The annual payment may be determined by use of the equation:

$$\text{annual payment} = (i/((1+i)^n - 1) + i) \times 12{,}000, \qquad (8\text{-}1)$$

where i is interest rate expressed as a fraction, i.e., 0.09. A longer term mortgage would decrease the annual payments as would a lower interest rate. If the load balance (principal) is reduced by each monthly payment, the above equation yields $100.70 per month by using a monthly interest rate of 0.75 percent and 300 monthly payments.

ENERGY SAVINGS FROM SOLAR ENERGY

The particular house had a design heating load of 24,000 Btu per ^{o}F-day. Fort Collins has an average of 6,000 ^{o}F-days. Consequently, the annual heat required is 144 million Btu per year. Energy input of 2.4×10^8 Btu per year would be required assuming a natural gas furnace with a thermal efficiency of 0.6.

Natural gas available in Fort Collins has a Btu content of 880 Btu per cubic foot, when a local gas rated at 1000 Btu per cubic foot at sea level is de-rated for the 5,000-foot elevation. Thus, the total gas required for heating this house with a conventional natural gas heating system is calculated at 273,000 cubic feet. An examination of current local rate schedules (1977) reveals the average 30,300 cubic feet required per month over a nine-month heating period would cost $38.96 per month plus $1.50 per month over the summer months, or $355.14 per year at current energy prices.

Solar energy will supply approximately 80 percent of the heating load; hence, it would save approximately $281 per heating season at current natural gas prices. Unfortunately, this $281 annual fuel saving is a long way from meeting the extra mortgage payment of $1,221.68 per year.

Alternative fuels such as LPG and electricity make solar energy more attractive due to their higher energy costs. If electricity were used to heat this particular house, the annual electric cost would be approximately $1,226 (at current electrical rates $0.03/kwh), of which 80 percent, $1,013, could be saved by the solar system. The saving is thus nearly enough to cover the annual mortgage payment for a solar system.

BREAKEVEN CALCULATIONS

Energy prices are certainly expected to rise in the future. Rather than attempt to predict the rise, one may find the breakeven point in terms of uniform annual cost for augmented solar systems versus conventional heating systems.

In order to determine the breakeven cost, one sets up an equation containing the uniform annual cost of augmented solar heating on one side of an equality and the costs of conventional heating on the other side, as shown below.

BREAKEVEN COST WITH GAS

SOLAR	CONVENTIONAL (GAS)
\$1,221.68/yr for mortgage + 20 percent of conventional fuel + electric pumping costs @ approximately 50 kwh/mo @ \$0.04/kwh	273,000 cubic feet of natural gas per year

$\$1,221,68/\text{yr} + (.20)\ (273,000\ \text{ft}^3/\text{yr})\ (\$X/\text{ft}^3) + \$2/\text{mo (elec.)} \times 12\ \text{mo/yr} =$

$273,000\ \text{ft}^3/\text{yr}\ (X/\text{ft}^3)$

$\$1,245.68 = (218,400\ \text{ft}^3/\text{yr})\ (\$X/\text{ft}^3)$

$\$1,245.68/\text{yr} = (218,400)\ (\$X/\text{ft}^3)$

$$\text{Breakeven} = \$0.0057/\text{ft}^3$$

$$= \$0.57/100\ \text{ft}^3$$

Thus solar heating would be economically competitive in this case if natural gas cost $\$0.57/100\ \text{ft}^3$. How long it will take for natural gas to reach that value from its present value of $\$0.10/100\ \text{ft}^3$ is anybody's guess. A rough estimate can be made if one assumes some annual percentage increases. A range of possible annual increases and the corresponding years required to double, triple, and quadruple is shown in the table below.

YEARS REQUIRED FOR A VALUE TO DOUBLE, TRIPLE, OR QUADRUPLE.
ANNUAL PERCENTAGE INCREASE

	5%	10%	15%	20%	25%
2X	14	7	5	4	3
3X	23	12	8	6	5
4X	28	15	10	8	6

The breakeven cost for electrical heating for this particular house is shown below.

BREAKEVEN COST WITH ELECTRICITY

SOLAR	CONVENTIONAL (ELECTRIC)
\$1,221.68/yr for mortgage + 20 percent of electric heating + electric pumping costs @ approximately \$2/mo	42,192 kwh/yr

$$(\$1,221.68 + \$2 \times 12) + (0.20 \times 42,192 \text{ kwh/yr})(\$X/\text{kwh}) =$$

$$42,192\ (42,192 \text{ kwh/yr})\ (\$X/\text{kwh})$$

$$\therefore\ 1,245.68 + 8,438\,X = 42,192\,X$$

$$33,754\,X = 1,245.68$$

$$\therefore\ X = \$0.0369/\text{kwh}$$

The above calculations indicate that natural gas prices in the Fort Collins area must increase by nearly 4½ times over their present values before a solar system would become economically competitive with a natural gas heating system. However, present electrical rates are almost at a level at which solar systems are economically competitive with electric resistance heating systems. The economics of solar systems as energy prices increase are investigated in the next section.

ECONOMICS OF SOLAR HEATING AS ENERGY COSTS INCREASE

YEARLY CASH FLOW COMPARISONS

The economic attractiveness of residential solar heating would, presumably, increase as conventional fuel prices increase. Table 8-3 shows the annual cash flow for a solar system with a gas-fired auxiliary system, with the solar system providing 80 percent of the annual heating load. Also presented is the annual cash flow for the conventional heating

Table 8-3. Yearly Cash Flow for Heating with Solar Energy and with Natural Gas

	Solar with Gas Auxiliary				Conventional Heating	
	(1)	(2)	(3) Electric Pumping	(4) 1+2+3	(5)	(6) (5)-(4)
Year	Mortgage	Gas @ 20% of Conventional	@ 7% Annual Increase	Total	Natural Gas @ 10% Annual Increase	Incremental Cash Flow
1	1222	54	24	1300	272	-1028
2	1222	60	26	1308	299	-1009
3	1222	66	27	1315	329	- 986
4	1222	72	29	1323	362	- 961
5	1222	80	31	1333	399	- 934
6	1222	88	34	1344	438	- 906
7	1222	96	36	1354	482	- 872
8	1222	106	39	1367	530	- 837
9	1222	117	41	1380	583	- 797
10	1222	128	44	1394	641	- 753
11	1222	141	47	1410	706	- 704
12	1222	155	51	1428	776	- 652
13	1222	171	54	1447	854	- 593
14	1222	188	58	1468	940	- 528
15	1222	207	62	1491	1034	- 457
16	1222	227	66	1515	1137	- 378
17	1222	250	71	1543	1251	- 292
18	1222	275	76	1573	1376	- 197
19	1222	303	81	1606	1513	- 93
20	1222	333	87	1642	1665	23
21	1222	366	93	1681	1831	150
22	1222	403	99	1724	2014	290
23	1222	443	106	1771	2216	445
24	1222	487	114	1823	2437	614
25	1222	536	122	1880	2681	801
				$37,420	$26,766	$-10,654

system and the incremental cash flow between the solar system and the conventional system. In developing this table, it was assumed that natural gas prices would increase at 10 percent per year and that electricity costs would increase at 7 percent per year. It is clear from this table that the solar system is not economically competitive with the gas-fired conventional system. This conclusion is changed dramatically when one considers an electrical auxiliary system. These results are shown in Table 8-4. The right-most column again presents the incremental cash flow of the solar system over the costs of a conventional system. The negative values in the early years indicate that the solar cost is greater than the conventional cost, but this situation reverses after five years. The total savings over the assumed 25-year lifetime of the system are significant for the solar system as compared with the conventional electric heating system.

PRESENT WORTH CALCULATIONS

The present worth of the operating costs of the solar system and the conventional heating costs, not including the mortgage payment for the solar system, has been calculated, and the results are presented in Tables 8-5 and 8-6. Table 8-5 presents results for a solar system with gas auxiliary, whereas Table 8-6 presents the results for a solar system with electric auxiliary. The future operating costs have been discounted to an equivalent present worth by assuming a cost of money of 9 percent per year, the same rate assumed for the mortgage. The present worth is given by

$$P = F(1/(1+i)^n) \qquad (8-2)$$

Table 8-4. Yearly Cash Flow for Space Heating with Solar Energy and with Electricity

	Solar and Electric Auxiliary				Conventional Heating	
	(1)	(2)	(3)	(4) 1+2+3	(5)	(6) (5)-(4)
Year	Mortgage	Electricity @ 20% of Conventional	Electric Pumping @ 7% Annual Increase	Total	Electric @ 7% Annual Increase	Incremental Cash Flow
1	1222	253	24	1499	1266	-233
2	1222	271	26	1519	1355	-164
3	1222	290	27	1539	1449	- 90
4	1222	310	29	1561	1551	- 10
5	1222	332	31	1585	1659	74
6	1222	355	34	1611	1776	165
7	1222	380	36	1638	1900	262
8	1222	406	39	1667	2033	366
9	1222	435	41	1698	2175	477
10	1222	465	44	1731	2327	596
11	1222	498	47	1767	2490	723
12	1222	533	51	1806	2665	859
13	1222	570	54	1846	2851	1005
14	1222	610	58	1890	3051	1161
15	1222	652	62	1936	3264	1328
16	1222	698	66	1986	3493	1507
17	1222	747	71	2040	3737	1697
18	1222	799	76	2097	3999	1902
19	1222	855	81	2158	4279	2121
20	1222	915	87	2224	4579	2355
21	1222	979	93	2294	4899	2605
22	1222	1048	99	2369	5242	2873
23	1222	1121	106	2449	5609	3160
24	1222	1199	114	2535	6002	3467
25	1222	1283	122	2627	6422	3795
				$48,072	$80,073	$32,001

Table 8-5. Present Worth of Solar and Gas System

	(1)	(2)	(3)	(4)	(5)
				Present Worth at 9% Discounting	
Year	Conventional Heating with 10% Annual Increase in Gas	Solar System Without Mortgage Payment	Present Worth Factor at 9%	Conventional	Solar
		(2)+(3) from Table 8-3		(3)x(1)	(3)x(2)
1	272	78	.9174	250	72
2	299	86	.8417	252	72
3	329	93	.7722	254	72
4	362	101	.7084	256	72
5	399	111	.6499	259	72
6	438	122	.5963	261	73
7	482	132	.5470	264	72
8	530	145	.5019	266	72
9	583	158	.4606	268	73
10	641	172	.4224	271	73
11	706	188	.3875	274	73
12	776	206	.3555	276	73
13	854	225	.3262	279	73
14	940	246	.2992	281	74
15	1,034	269	.2745	284	74
16	1,137	293	.2519	286	74
17	1,251	321	.2311	289	74
18	1,376	351	.2120	292	74
19	1,513	384	.1945	294	75
20	1,665	420	.1784	297	75
21	1,831	459	.1637	300	75
22	2,014	502	.1502	303	75
23	2,216	549	.1378	305	76
24	2,437	601	.1264	308	76
25	2,681	658	.1160	311	76
	$26,766	$6,870		$6,980	$1,840

Economic Maximum First Cost
for Solar = $6,980 - $1,840 = $5,140

Table 8-6. Present Worth of Solar and Electric System

Year	(1) Conventional Heating with 7% Annual Increase in Electricity	(2) Solar System Without Mortgage Payments	(3) Present Worth Factor at 9%	(4) Present Worth at 9% Discounting: Conventional	(5) Solar
		(2)+(3) from Table 8-4		(3)x(1)	(3)x(2)
1	1,266	277	.9174	1,161	254
2	1,355	297	.8417	1,141	250
3	1,449	317	.7722	1,119	245
4	1,551	339	.7084	1,099	240
5	1,659	363	.6499	1,078	236
6	1,776	389	.5963	1,059	232
7	1,900	416	.5470	1,039	228
8	2,033	445	.5019	1,020	223
9	2,175	476	.4604	1,001	219
10	2,327	509	.4224	983	215
11	2,490	545	.4875	965	266
12	2,665	584	.3555	947	208
13	2,851	624	.3262	930	204
14	3,051	668	.2992	913	200
15	3,264	714	.2745	896	196
16	3,493	764	.2519	880	192
17	3,737	818	.2311	864	189
18	3,999	875	.2120	848	186
19	4,279	936	.1945	832	182
20	4,579	1,002	.1784	817	179
21	4,899	1,072	.1637	802	175
22	5,242	1,147	.1502	787	172
23	5,609	1,227	.1378	778	169
24	6,002	1,313	.1264	759	166
25	6,422	1,405	.1160	745	163
	$80,073	$17,522		$23,463	$5,189

Economic Maximum First Cost
for Solar = $23,463 - $5,189 = $18,274

where P represents the present worth and F represents the future value.

The present worth of the future cash flows of the conventional heating systems is $6,980, compared with $1,840 for the solar system without mortgage payment. The difference of $5,140 represents the economic maximum first cost of solar component[illegible]s house if money is worth 9 percent per year, and if natural gas prices rise at a 10 percent compounded annual rate. The results are much more favorable for solar when compared with electric auxiliary as shown in Table 8-6. In this case, we see that the economic maximum first cost for solar exceeds by approximately $6,000 the actual cost of the solar system.

There are other factors that one could consider in performing these economic analyses. For example, the interest paid on the mortgage could be used to reduce the homeowner's personal income tax. However, one should also consider that property taxes, insurances, and the cost of the floor space occupied by the solar system (primarily the heat storage unit) should be included in the solar system costs, partially offsetting the savings realized from income tax reductions. For commercial type installations, depreciation of the solar system would also represent a favorable tax situation. Regardless of what manipulations one might perform, the fact remains that, at the present time, solar systems are not economically competitive with gas systems in this region, but they are now competitive with electrical systems in this region. For geographical regions other than Fort Collins, this situation does not necessarily prevail. Using the techniques presented above, one can readily calculate the economics of solar systems for any region and any inflation rates.

The economic maximum first cost for a solar system may be determined very quickly by reference to Tables 8-7 through 8-27[1]. These tables were developed for mortgage periods of 20, 25, and 30 years, interest rates between 8 and 12 percent, and annual increases in conventional fuel costs between 0 and 12 percent. The use of the tables will be illustrated by the example shown in Table 8-4. The mortgage period was assumed to be 25 years, the mortgage interest rate was 9 percent, and it was assumed that electricity costs would increase at 7 percent per year. The cost of heating with electricity during the first year was determined to be $1,266. The solar system was designed to provide 80 percent of the heating requirements.

The factor F, shown in Tables 8-7 through 8-27, is defined as

$$F = \frac{\text{First Year's Fuel and Operating Cost}}{\text{First Year's Conventional Fuel Cost}}$$

Therefore, in this example,

$$F = \frac{253 + 24}{1,266} = .22$$

We will have to interpolate since there is no table for the assumed annual rate of fuel increase of 7 percent. From Tables 8-17 and 8-18 we see that the cost factor is 13.0593 for 6-percent annual rate of fuel increase and 16.0522 for 8-percent annual rate of fuel increase. The economic maximum first cost for solar is determined by multiplying the fuel cost factor by the first year's conventional fuel cost.

---

[1] S. E. Huck, "Design Charts for Solar Heating Systems," M.S. Thesis, Department of Mechanical Engineering, Colorado State University, Fort Collins, Colorado 80523 (1976).

Therefore, we obtain

$$\begin{aligned} EMFC &= (\$1,266)(13.0593) \\ &= \$16,533 \qquad (6\%\ ARI) \end{aligned}$$

and

$$\begin{aligned} EMFC &= (\$1,266)(16.0522) \\ &= \$20,322 \qquad (8\%\ ARI) \end{aligned}$$

we need to take one-half the difference between these two values and add this to $16,533 to determine the economic maximum first cost for solar for the assumed 7-percent annual rate of increase in fuel costs. We obtain

$$\begin{aligned} EMFC &= \$16,533 + 0.5(20,322 - 16,533) \\ &= \$18,427 . \end{aligned}$$

The detailed analysis in Table 8-6 resulted in a figure of $18,274. The slight difference is due to round-off in our calculations.

The cost estimate for a given solar system should be compared with the economic maximum first cost for a solar system in order to determine if the solar system should be installed. If EMFC exceeds the estimated cost, then it would be economically advantageous for the owner to install the solar system. Typically, a curve such as that shown in Figure 8-1 should be developed. The two curves shown in this figure represent the economic maximum first cost and the estimated cost as functions of the collector area. The points of intersection represent breakeven points. That is, the solar system and the conventional system would have equal costs over the lifetime of the mortgage. The shaded region between the two curves represents a region in which the solar system would have an

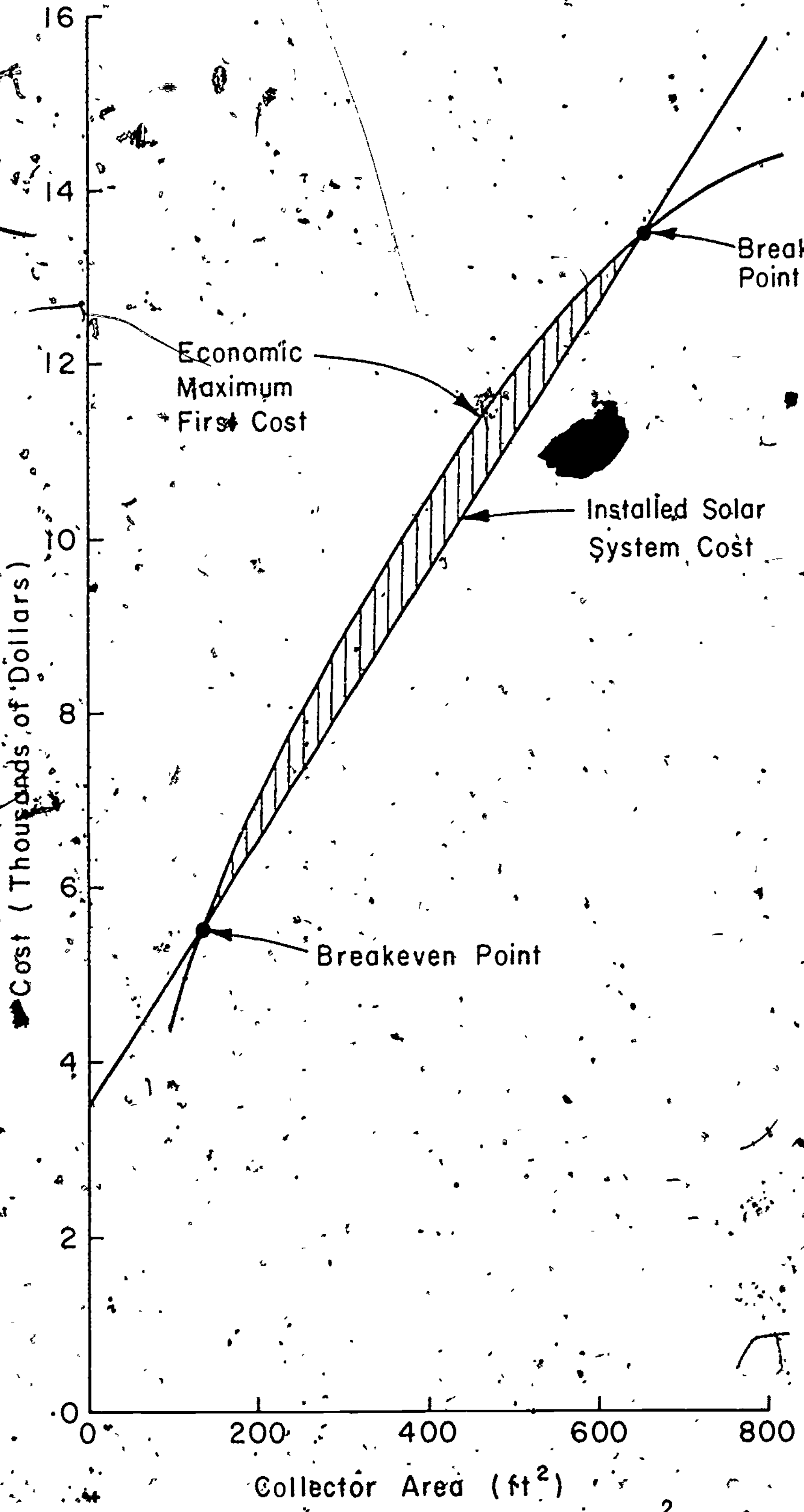

Figure 8-1. Cost Versus Area[2]

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[2] C. B. Winn and G. R. Johnson, "Solar Energy Analysis Programs for Programmable Handheld Calculators," Report No. TR-99, SEEC, Inc., Fort Collins, Colorado, November 1976.

economic advantage over the conventional system. The optimal collector area would be at the point of maximum separation between the two curves. In this case that would be at approximately 400 square feet.

ALTERNATE APPROACHES

There are several alternate approaches to performing economic analyses, but most lead to essentially the same conclusions, depending, of course, upon what assumptions are made. One of the alternative methods is an analysis procedure programmed into the interactive computer program developed at the University of Wisconsin.

If desired, the program will perform a life-cycle cost economic analysis which compares the costs of the solar-assisted system with the costs of the conventional system on a present-value basis. The program estimates the timing and amounts of annual cash flows using the following equations:

$$\underset{\text{with solar}}{\text{yearly cost}} = \underset{\text{payment}}{\text{mortgage}} + \underset{\text{fuel costs}}{\text{backup system}} + \underset{\text{costs}}{\text{misc.}} + \underset{\text{increase}}{\text{property tax}} - \underset{\text{with solar}}{\underset{\text{decrease}}{\text{income tax}}}$$

$$\underset{\text{w/o solar}}{\text{yearly cost}} = \underset{\text{fuel cost}}{\text{conventional system}} - \underset{\text{w/o solar}}{\underset{\text{decrease}}{\text{income tax}}}$$

The terms are defined in the following. It is assumed that the solar backup system is identical to the conventional heating and domestic water system. Therefore, only the additional investment due to the solar system need be considered in the analysis.

USE OPTIMIZED COLLECTOR AREA = 1, SPECIFIED AREA = 2

The user specifies a collector area via system parameter number 2. However, if an economically optimized collector area is desired, the program ignores the specified collector area and performs a numerical search for an optimum area. The criterion used is to find the collector area which minimized the present value of all of the yearly costs of the solar-assisted system over the period of analysis.

PERIOD OF THE ECONOMIC ANALYSIS

This specifies the number of years over which the life-cycle cost analysis will be performed.

COLLECTOR AREA DEPENDENT COSTS

Some of the extra costs of a solar heating system above the conventional system are collector area dependent. These include the cost of storage and of the collector.

CONSTANT SOLAR COSTS

This refers to extra costs of solar heating systems above the conventional system which are not dependent on the collector size. Examples are costs for architectural modifications, piping or ducts, controls, pumps or blowers, etc.

DOWN PAYMENT (% OF ORIGINAL INVESTMENT)

The original investment refers to the extra investment required to put in the solar system. Therefore, the percentage which is paid down on the solar system equals the ratio of the incremental increase in the down payment required by the lender to the incremental increase in the size of the loan required due to the solar system.

ANNUAL INTEREST RATE ON MORTGAGE

This is the annual interest rate charged by your lender.

TERM OF THE MORTGAGE

The number of years over which you must pay off the loan.

ANNUAL NOMINAL (MARKET) DISCOUNT RATE

This refers to the annual rate of return which you make with your money in your best investment opportunity. The annual nominal or market rate of return equals the real rate of return plus the general inflation rate. For the typical homeowner, the real rate of return is 1-2 percent; for business, 3-4 percent.

EXPENSES (INSURANCE, MAINTENANCE) OF SYSTEM IN FIRST YEAR

All additional yearly expenses due to the solar system which cannot be input anywhere else should be included here.

ANNUAL INCREASE IN ABOVE EXPENSES

Allowance can be made for the annual rate of increase of insurance and maintenance costs (i.e., the general inflation rate) via this parameter.

PRESENT COST OF AUXILIARY FUEL (CF)

This is the actual present cost of the backup system fuel, times 100, divided by the efficiency of the backup system heating unit. The actual present cost of the fuel should include any fuel adjustment charges beyond the standard rate.

CF RISE: LINEAR = 1, %/yr = 2, SEQUENCE OF VALUES = 3

The program user may allow fuel costs to rise in any of three possible ways so that any scenario can be investigated. These increases should include general inflation plus any net increases in fuel costs.

IF 1, WHAT IS THE SLOPE OF CF INCREASE?

If a linear fuel cost rise is assumed, the slope of increase is required. Otherwise this parameter is ignored.

IF 2, WHAT IS THE ANNUAL RATE OF CF RISE?

If a %/yr fuel cost rise is desired, the annual rate of fuel cost increase must be input here. Otherwise this parameter is ignored.

ECONOMIC PRINTOUT BY YEAR = 1, CUMULATIVE = 2

If a yearly printout is desired, several cash flows are printed each year of the economic analysis. If a cumulative printout is desired, the present value of the yearly costs over the period are output for the building with and without a solar energy system.

EFFECTIVE FEDERAL-STATE INCOME TAX RATE

State income taxes paid are deductible on federal returns; therefore, the effective federal-state income tax rate is calculated as

Effective Rate = Federal Rate + State Rate - (Federal Rate) x (State Rate)

TRUE PROPERTY TAX RATE PER $ OF ORIGINAL INVESTMENT

Property tax rates are applied to your assessed value. Therefore, an estimate of assessed value as a percentage of original investment is required so that

$$\frac{\text{Tax Rate}}{\text{\$ Original Invest.}} = \left(\frac{\text{Tax Rate}}{\text{Assessed Value}}\right) \times \left(\frac{\text{Assessed Value}}{\text{Original Invest.}}\right)$$

INCOME-PRODUCING BUILDING? YES = 1, NO = 2

The economic analysis for commercial and residential buildings is different because businesses benefit from more income tax deductions due to the added investment required for the solar heating system. For the homeowner, interest and property taxes paid are deductible on income taxes. For a business, interest, depreciation, fuel expenses, property taxes and maintenance and insurance costs are all deductible.

If your building is not income-producing and does not qualify as a business investment, the remaining parameters are ignored.

The yearly cost equations were given earlier. For a non-income-producing building, such as a residence, the income tax terms are given below.

$$\begin{array}{l}\text{income tax}\\ \text{decrease}\\ \text{with solar}\end{array} = \begin{array}{l}\text{tax}\\ \text{rate}\end{array} \times (\text{interest paid} + \text{property tax paid})$$

$$\begin{array}{l}\text{income tax}\\ \text{decrease}\\ \text{w/o solar}\end{array} = 0$$

However, for a commercial building the income tax terms are as follows:

$$\begin{array}{l}\text{income tax}\\ \text{decrease}\\ \text{with solar}\end{array} = \begin{array}{l}\text{tax}\\ \text{rate}\end{array} \times \left(\begin{array}{l}\text{interest}\\ \text{paid}\end{array} + \begin{array}{l}\text{property}\\ \text{tax}\\ \text{paid}\end{array} + \begin{array}{l}\text{misc.}\\ \text{expense}\end{array} + \begin{array}{l}\text{backup system}\\ \text{fuel cost}\end{array} + \text{depreciation}\right)$$

income tax = tax rate x (conventional system fuel costs)
decrease
w/o solar

Although commercial building owners have more income tax deductions due to the solar investment than homeowners, they do not save as much in fuel costs since fuel costs are deductible whether or not they install a solar heating system.

Keep in mind that the interest, property tax, miscellaneous expense, and depreciation deductions refer only to the incremental increase in these deductions due to the solar investment. Therefore, these terms do not appear in the "without solar" income tax decrease equation.

DEPRECIATION: STRAIGHT LINE = 1, DECLINING BALANCE = 2, SUM-OF-YEARS - DIGITS = 3, NONE = 4

Any of the standard methods of depreciation can be used. Depreciation deductions due to the extra investment due to solar are calculated in order to estimate the income tax savings.

IF 2, WHAT PERCENT OF STRAIGHT-LINE DEPRECIATION RATE IS DESIRED?

The federal government allows several rates at which investments can be written off using the declining balance method. These rates are expressed as the percent of straight-line depreciation rate allowed.

USEFUL LIFE FOR DEPRECIATION PURPOSES

This is the length of time over which you intend to depreciate out your investment.

SALVAGE VALUE AT END OF DEPRECIATION PERIOD

An estimate of the system's salvage value at the end of the depreciation period is required in order to calculate depreciation.

This procedure will be illustrated during the computation sessions using the interactive computer terminal.

Table 8-7. Fuel Cost Factors

ANNUAL RATE OF FUEL INCREASE =0.00
NUMBER OF YEARS = 20

I	0.08	0.09	0.10	0.11	0.12
F	----	----	----	----	----
.10	8.8363	8.2157	7.6622	7.1670	6.7225
.12	8.6400	8.0331	7.4919	7.0077	6.5731
.14	8.4436	7.8505	7.3217	6.8485	6.4237
.16	8.2472	7.6680	7.1514	6.6892	6.2743
.18	8.0509	7.4854	6.9811	6.5299	6.1249
.20	7.8545	7.3028	6.8109	6.3707	5.9756
.22	7.6582	7.1203	6.6406	6.2114	5.8262
.24	7.4618	6.9377	6.4703	6.0521	5.6768
.26	7.2654	6.7551	6.3000	5.8929	5.5274
.28	7.0691	6.5726	6.1298	5.7336	5.3780
.30	6.8727	6.3900	5.9595	5.5743	5.2286
.32	6.6763	6.2074	5.7892	5.4151	5.0792
.34	6.4800	6.0248	5.6190	5.2558	4.9298
.36	6.2836	5.8423	5.4487	5.0965	4.7804
.38	6.0873	5.6597	5.2784	4.9373	4.6311
.40	5.8909	5.4771	5.1081	4.7780	4.4817
.42	5.6945	5.2946	4.9379	4.6187	4.3323
.44	5.4982	5.1120	4.7676	4.4595	4.1829
.46	5.3018	4.9294	4.5973	4.3002	4.0335
.48	5.1054	4.7468	4.4271	4.1409	3.8841
.50	4.9091	4.5643	4.2568	3.9817	3.7347
.52	4.7127	4.3817	4.0865	3.8224	3.5853
.54	4.5163	4.1991	3.9162	3.6631	3.4359
.56	4.3200	4.0166	3.7460	3.5039	3.2866
.58	4.1236	3.8340	3.5757	3.3446	3.1372
.60	3.9273	3.6514	3.4054	3.1853	2.9878
.62	3.7309	3.4688	3.2352	3.0261	2.8384
.64	3.5345	3.2863	3.0649	2.8668	2.6890
.66	3.3382	3.1037	2.8946	2.7075	2.5396
.68	3.1418	2.9211	2.7243	2.5483	2.3902
.70	2.9454	2.7386	2.5541	2.3890	2.2408
.72	2.7491	2.5560	2.3838	2.2297	2.0914
.74	2.5527	2.3734	2.2135	2.0705	1.9421
.76	2.3564	2.1909	2.0433	1.9112	1.7927
.78	2.1600	2.0083	1.8730	1.7519	1.6433
.80	1.9636	1.8257	1.7027	1.5927	1.4939
.82	1.7673	1.6431	1.5324	1.4334	1.3445
.84	1.5709	1.4606	1.3622	1.2741	1.1951
.86	1.3745	1.2780	1.1919	1.1149	1.0457
.88	1.1782	1.0954	1.0216	.9556	.8963
.90	.9818	.9129	.8514	.7963	.7469

Table 8-8. Fuel Cost Factors

ANNUAL RATE OF FUEL INCREASE = .02
NUMBER OF YEARS = 20

I F	0.08	0.09	0.10	0.11	0.12
.10	10.2179	9.4482	8.7651	8.1569	7.6136
.12	9.9908	9.2383	8.5704	7.9757	7.4444
.14	9.7638	9.0283	8.3756	7.7944	7.2752
.16	9.5367	8.8183	8.1808	7.6131	7.1060
.18	9.3096	8.6084	7.9860	7.4319	6.9368
.20	9.0826	8.3984	7.7912	7.2506	6.7677
.22	8.8555	8.1885	7.5965	7.0693	6.5985
.24	8.6284	7.9785	7.4017	6.8881	6.4293
.26	8.4014	7.7685	7.2069	6.7068	6.2601
.28	8.1743	7.5586	7.0121	6.5255	6.0909
.30	7.9472	7.3486	6.8173	6.3443	5.9217
.32	7.7202	7.1387	6.6225	6.1630	5.7525
.34	7.4931	6.9287	6.4278	5.9817	5.5833
.36	7.2661	6.7187	6.2330	5.8005	5.4141
.38	7.0390	6.5088	6.0382	5.6192	5.2449
.40	6.8119	6.2988	5.8434	5.4379	5.0757
.42	6.5849	6.0888	5.6486	5.2567	4.9065
.44	6.3578	5.8789	5.4539	5.0754	4.7374
.46	6.1307	5.6689	5.2591	4.8942	4.5682
.48	5.9037	5.4590	5.0643	4.7129	4.3990
.50	5.6766	5.2490	4.8695	4.5316	4.2298
.52	5.4495	5.0390	4.6747	4.3504	4.0606
.54	5.2225	4.8291	4.4800	4.1691	3.8914
.56	4.9954	4.6191	4.2852	3.9878	3.7222
.58	4.7683	4.4092	4.0904	3.8066	3.5530
.60	4.5413	4.1992	3.8956	3.6253	3.3838
.62	4.3142	3.9892	3.7008	3.4440	3.2146
.64	4.0872	3.7793	3.5061	3.2628	3.0454
.66	3.8601	3.5693	3.3113	3.0815	2.8763
.68	3.6330	3.3594	3.1165	2.9002	2.7071
.70	3.4060	3.1494	2.9217	2.7190	2.5379
.72	3.1789	2.9394	2.7269	2.5377	2.3687
.74	2.9518	2.7295	2.5322	2.3564	2.1995
.76	2.7248	2.5195	2.3374	2.1752	2.0303
.78	2.4977	2.3096	2.1426	1.9939	1.8611
.80	2.2706	2.0996	1.9478	1.8126	1.6919
.82	2.0436	1.8896	1.7530	1.6314	1.5227
.84	1.8165	1.6797	1.5582	1.4501	1.3535
.86	1.5894	1.4697	1.3635	1.2689	1.1843
.88	1.3624	1.2598	1.1687	1.0876	1.0151
.90	1.1353	1.0498	.9739	.9063	.8460

Table 8-9. Fuel Cost Factors

ANNUAL RATE OF FUEL INCREASE = .04
NUMBER OF YEARS = 20

I	0.08	0.09	0.10	0.11	0.12
F					
.10	11.9227	10.8626	10.1146	9.3629	8.6946
.12	11.6578	10.7190	9.8898	9.1548	8.5014
.14	11.3928	10.4754	9.6650	8.9468	8.308[illegible]
.16	11.1279	10.2318	9.4402	8.7387	8.115[illegible]
.18	10.8629	9.9882	9.2155	8.5307	7.9217
.20	10.5980	9.7446	8.9907	8.3226	7.7285
.22	10.3330	9.5010	8.7659	8.1145	7.5353
.24	10.0681	9.2573	8.5412	7.9065	7.3421
.26	9.8031	9.0137	8.3164	7.6984	7.1489
.28	9.5382	8.7701	8.0916	7.4903	6.9557
.30	9.2732	8.5265	7.8669	7.2823	6.7625
.32	9.0083	8.2829	7.6421	7.0742	6.5693
.34	8.7433	8.0393	7.4173	6.8661	6.3760
.36	8.4784	7.7957	7.1926	6.6581	6.1828
.38	8.2134	7.5520	6.9678	6.4500	5.9896
.40	7.9485	7.3084	6.7430	6.2419	5.7964
.42	7.6835	7.0648	6.5183	6.0339	5.6032
.44	7.4186	6.8212	6.2935	5.8258	5.4100
.46	7.1536	6.5776	6.0687	5.6177	5.2168
.48	6.8887	6.3340	5.8440	5.4097	5.0235
.50	6.6237	6.0304	5.6192	5.2016	4.8303
.52	6.3588	5.8467	5.3944	4.9936	4.6371
.54	6.0938	5.6031	5.1697	4.7855	4.4439
.56	5.8289	5.3595	4.9449	4.5774	4.2507
.58	5.5639	5.1159	4.7201	4.3694	4.0575
.60	5.2990	4.8723	4.4954	4.1613	3.8643
.62	5.0340	4.6287	4.2706	3.9532	3.6711
.64	4.7691	4.3851	4.0458	3.7452	3.4778
.66	4.5041	4.1414	3.8211	3.5371	3.2846
.68	4.2392	3.8978	3.5963	3.3290	3.0914
.70	3.9742	3.6542	3.3715	3.1210	2.8982
.72	3.7093	3.4106	3.1467	2.9129	2.7050
.74	3.4443	3.1670	2.9220	2.7048	2.5118
~~.76~~	3.1794	2.9234	2.6972	2.4968	2.3186
.78	2.9144	2.6798	2.4724	2.2887	2.1253
.80	2.6495	2.4361	2.2477	2.0806	1.9321
.82	2.3845	2.1925	2.0229	1.8726	1.7389
.84	2.1196	1.9489	1.7981	1.6645	1.5457
.86	1.8546	1.7053	1.5734	1.4565	1.3525
.88	1.5897	1.4617	1.3486	1.2484	1.1593
.90	1.3247	1.2181	1.1238	1.0403	.9661

Table 8-10. Fuel Cost Factors

ANNUAL RATE OF FUEL INCREASE = .06
NUMBER OF YEARS = 20

I	0.08	0.09	0.10	0.11	0.12
F					
.10	14.0362	12.8324	11.7738	10.8397	10.0129
.12	13.[illegible]243	12.5473	11.5122	10.5988	9.7904
.14	13.4123	12.2621	11.2505	10.3580	9.5679
.16	13.1004	11.9769	10.9889	10.1171	9.3454
.18	12.7885	11.6918	10.7272	9.8762	9.1229
.20	12.4766	11.4066	10.4656	9.6353	8.9004
.22	12.1647	11.1214	10.2040	9.3944	8.6778
.24	11.8528	10.8363	9.9423	9.1535	8.4553
.26	11.5408	10.5511	9.6807	8.9127	8.2328
.28	11.2289	10.2660	9.4190	8.6718	8.0103
.30	10.9170	9.9808	9.1574	8.4309	7.7878
.32	10.6051	9.6956	8.8958	8.1900	7.5653
.34	10.2932	9.4105	8.6341	7.9491	7.3428
.36	9.9813	9.1253	8.3725	7.7082	7.1203
.38	9.6694	8.8401	8.1108	7.4674	6.8978
.40	9.3574	8.5550	7.8492	7.2265	6.6753
.42	9.0455	8.2698	7.5876	6.9856	6.4528
.44	8.7336	7.9846	7.3259	6.7447	6.2302
.46	8.4217	7.6995	7.0643	6.5038	6.0077
.48	8.1098	7.4143	6.8026	6.2629	5.7852
.50	7.7979	7.1291	6.5410	6.0221	5.5627
.52	7.4860	6.8440	6.2794	5.7812	5.3402
.54	7.1740	6.5588	6.0177	5.5403	5.1177
.56	6.8621	6.2736	5.7561	5.2994	4.8952
.58	6.5502	5.9885	5.4944	5.0585	4.6727
.60	6.2383	5.7033	5.2328	4.8177	4.[illegible]502
.62	5.9264	5.4181	4.9712	4.5768	4.2277
.64	5.6145	5.1330	4.7095	4.3359	4.0052
.66	5.3026	4.8478	4.4479	4.0950	3.7827
.68	4.9906	4.5626	4.1862	3.8541	3.5601
.70	4.6787	4.2775	3.9246	3.6132	3.3376
.72	4.3668	3.9923	3.6630	3.3724	3.1151
.74	4.0549	3.7071	3.[illegible]	3.1315	2.8926
.76	3.7430	3.4220	3.1397	2.8906	2.6701
.78	3.4311	3.1368	2.8780	2.6497	2.4476
.80	3.1191	2.8517	2.6164	2.4088	2.2251
.82	2.8072	2.5665	2.3548	2.1679	2.0026
.84	2.4953	2.2813	2.0931	1.9271	1.7801
.86	2.1834	1.9962	1.8315	1.6862	1.5576
.88	1.8715	1.7110	1.5698	1.4453	1.3351
.90	1.5596	1.4258	1.3082	1.2044	1.1125

Table 8-11. Fuel Cost Factors

ANNUAL RATE OF FUEL INCREASE = .08
NUMBER OF YEARS = 20

F \ I	0.08	0.09	0.10	0.11	0.12
.10	16.6667	15.1507	13.8230	12.6565	11.6283
.12	16.2963	14.8140	13.5159	12.3752	11.3699
.14	15.9259	14.4773	13.2087	12.0940	11.1115
.16	15.5556	14.1407	12.9015	11.8127	10.8531
.18	15.1852	13.8040	12.5943	11.5315	10.5947
.20	14.8148	13.4673	12.2871	11.2502	10.3363
.22	14.4444	13.1306	11.9800	10.9690	10.0779
.24	14.0741	12.7939	11.6728	10.6877	9.8195
.26	13.7037	12.4573	11.3656	10.4065	9.5611
.28	13.3333	12.1206	11.0584	10.1252	9.3026
.30	12.9630	11.7839	10.7512	9.8439	9.0442
.32	12.5926	11.4472	10.4441	9.5627	8.7858
.34	12.2222	11.1105	10.1369	9.2814	8.5274
.36	11.8519	10.7738	9.8297	9.0002	8.2690
.38	11.4815	10.4372	9.5225	8.7189	8.0106
.40	11.1111	10.1005	9.2154	8.4377	7.7522
.42	10.7407	9.7638	8.9082	8.1564	7.4938
.44	10.3704	9.4271	8.6010	7.8752	7.2354
.46	10.0000	9.0904	8.2938	7.5939	6.9770
.48	9.6296	8.7537	7.9866	7.3126	6.7186
.50	9.2593	8.4171	7.6795	7.0314	6.4602
.52	8.8889	8.0804	7.3723	6.7501	6.2018
.54	8.5185	7.7437	7.0651	6.4689	5.9434
.56	8.1481	7.4070	6.7579	6.1876	5.6850
.58	7.7778	7.0703	6.4507	5.9064	5.4265
.60	7.4074	6.7337	6.1436	5.6251	5.1681
.62	7.0370	6.3970	5.8364	5.3439	4.9097
.64	6.6667	6.0603	5.5292	5.0626	4.6513
.66	6.2963	5.7236	5.2220	4.7813	4.3929
.68	5.9259	5.3869	4.9149	4.5001	4.1345
.70	5.5556	5.0502	4.6077	4.2188	3.8761
.72	5.1852	4.7136	4.3005	3.9376	3.6177
.74	4.8148	4.3769	3.9933	3.6563	3.3593
.76	4.4444	4.0402	3.6861	3.3751	3.1009
.78	4.0741	3.7035	3.3790	3.0938	2.8425
.80	3.7037	3.3668	3.0718	2.8126	2.5841
.82	3.3333	3.0301	2.7646	2.5313	2.3257
.84	2.9630	2.6935	2.4574	2.2500	2.0673
.86	2.5926	2.3568	2.1502	1.9688	1.8088
.88	2.2222	2.0201	1.8431	1.687[illegible]	1.5504
.90	1.8519	1.6834	1.5359	1.4063	1.2920

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Table 8-12. Fuel Cost Factors

ANNUAL RATE OF FUEL INCREASE = .10
NUMBER OF YEARS = 20

F \ I	0.08	0.09	0.10	0.11	0.12
.10	19.9518	18.0354	16.3636	14.9006	13.6162
.12	19.5084	17.6347	16.0000	14.5694	13.3136
.14	19.0650	17.2339	15.6364	14.2383	13.0110
.16	18.6217	16.8331	15.2727	13.9072	12.7084
.18	18.1783	16.4323	14.9091	13.5761	12.4059
.20	17.7349	16.0315	14.5455	13.2450	12.1033
.22	17.2915	15.6307	14.1818	12.9138	11.8007
.24	16.8482	15.2299	13.8182	12.5827	11.4981
.26	16.4048	14.8291	13.4545	12.2516	11.1955
.28	15.9614	14.4284	13.0909	11.9205	10.8929
.30	15.5181	14.0276	12.7273	11.5893	10.5904
.32	15.0747	13.6268	12.3636	11.2582	10.2878
.34	14.6313	13.2260	12.0000	10.9271	9.9852
.36	14.1879	12.8252	11.6364	10.5960	9.6826
.38	13.7446	12.4244	11.2727	10.2648	9.3800
.40	13.3012	12.0236	10.9091	9.9337	9.0775
.42	12.8578	11.6228	10.5455	9.6026	8.7749
.44	12.4144	11.2221	10.1818	9.2715	8.4723
.46	11.9711	10.8213	9.8182	8.9403	8.1697
.48	11.5277	10.4205	9.4545	8.6092	7.8671
.50	11.0843	10.0197	9.0909	8.2781	7.5645
.52	10.6410	9.6189	8.7273	7.9470	7.2620
.54	10.1976	9.2181	8.3636	7.6158	6.9594
.56	9.7542	8.8173	8.0000	7.2847	6.6568
.58	9.3108	8.4165	7.6364	6.9536	6.3542
.60	8.8675	8.0158	7.2727	6.6225	6.0516
.62	8.4241	7.6150	6.9091	6.2914	5.7491
.64	7.9807	7.2142	6.5455	5.9602	5.4465
.66	7.5373	6.8134	6.1818	5.6291	5.1439
.68	7.0940	6.4126	5.8182	5.2980	4.8413
.70	6.6506	6.0118	5.4545	4.9669	4.5387
.72	6.2072	5.6110	5.0909	4.6357	4.2361
.74	5.7638	5.2102	4.7273	4.3046	3.9336
.76	5.3205	4.8095	4.3636	3.9735	3.6310
.78	4.8771	4.4087	4.0000	3.6424	3.3284
.80	4.4337	4.0079	3.6364	3.3112	3.0258
.82	3.9904	3.6071	3.2727	2.9801	2.7232
.84	3.5470	3.2063	2.9091	2.6490	2.4207
.86	3.1036	2.8055	2.5455	2.3179	2.1181
.88	2.6602	2.4047	2.1818	1.9867	1.8155
.90	2.2169	2.0039	1.8182	1.6556	1.5129

Table 8-13. Fuel Cost Factors

ANNUAL RATE OF FUEL INCREASE = .12
NUMBER OF YEARS = 20

F \ I	0.08	0.09	0.10	0.11	0.12
.10	24.0659	21.6359	19.5237	17.6821	16.0714
.12	23.5311	21.1551	19.0898	17.2891	15.7143
.14	22.9963	20.6743	18.6560	16.8962	15.3571
.16	22.4615	20.1935	18.2221	16.5033	15.0000
.18	21.9267	19.7127	17.7883	16.1103	14.6429
.20	21.3919	19.2319	17.3544	15.7174	14.2857
.22	20.8571	18.7511	16.9205	15.3245	13.9286
.24	20.3223	18.2703	16.4867	14.9315	13.5714
.26	19.7875	17.7895	16.0528	14.5386	13.2143
.28	19.2527	17.3087	15.6190	14.1457	12.8571
.30	18.7179	16.8279	15.1851	13.7527	12.5000
.32	18.1831	16.3471	14.7512	13.3598	12.1429
.34	17.6483	15.8663	14.3174	12.9668	11.7857
.36	17.1135	15.3855	13.8835	12.5739	11.4286
.38	16.5787	14.9047	13.4497	12.1810	11.0714
.40	16.0439	14.4239	13.0158	11.7880	10.7143
.42	15.5091	13.9431	12.5819	11.3951	10.3571
.44	14.9743	13.4623	12.1481	11.0022	10.0000
.46	14.4395	12.9815	11.7142	10.6092	9.6429
.48	13.9047	12.5007	11.2804	10.2163	9.2857
.50	13.3699	12.0199	10.8465	9.8234	8.9286
.52	12.8351	11.5391	10.4126	9.4304	8.5714
.54	12.3003	11.0583	9.9788	9.0375	8.2143
.56	11.7655	10.5776	9.5449	8.6446	7.8571
.58	11.2307	10.0968	9.1111	8.2516	7.5000
.60	10.6959	9.6160	8.6772	7.8587	7.1429
.62	10.1612	9.1352	8.2433	7.4658	6.7857
.64	9.6264	8.6544	7.8095	7.0728	6.4286
.66	9.0916	8.1736	7.3756	6.6799	6.0714
.68	8.5568	7.6928	6.9418	6.2870	5.7143
.70	8.0220	7.2120	6.5079	5.8940	5.3571
.72	7.4872	6.7312	6.0740	5.5011	5.0000
.74	6.9524	6.2504	5.6402	5.1082	4.6429
.76	6.4176	5.7696	5.2063	4.7152	4.2857
.78	5.8828	5.2889	4.7725	4.3223	3.9286
.80	5.3480	4.8080	4.3386	3.9293	3.5714
.82	4.8132	4.3272	3.9047	3.5364	3.2143
.84	4.2784	3.8464	3.4709	3.1435	2.8571
.86	3.7436	3.3656	3.0370	2.7505	2.5000
.88	3.2088	2.8848	2.6032	2.3576	2.1429
.90	2.6740	2.4040	2.1693	1.9647	1.7857

Table 8-14. Fuel Cost Factors

ANNUAL RATE OF FUEL INCREASE =0.00
NUMBER OF YEARS = 25

F \ I	0.08	0.09	0.10	0.11	0.12
.10	9.6073	8.8403	8.1693	7.5796	7.0588
.12	9.3938	8.6439	7.9878	7.4111	6.9020
.14	9.1803	8.4474	7.8063	7.2427	6.7451
.16	8.9668	8.2510	7.6247	7.0743	6.5882
.18	8.7533	8.0545	7.4432	6.9058	6.4314
.20	8.5398	7.8581	7.2616	6.7374	6.2745
.22	8.3263	7.6616	7.0801	6.5690	6.1176
.24	8.1128	7.4652	6.8986	6.4005	5.9608
.26	7.8993	7.2687	6.7170	6.2321	5.8039
.28	7.6858	7.0723	6.5355	6.0637	5.6471
.30	7.4723	6.8758	6.3539	5.8952	5.4902
.32	7.2588	6.6794	6.1724	5.7268	5.3333
.34	7.0454	6.4829	5.9908	5.5584	5.1765
.36	6.8319	6.2865	5.8093	5.3899	5.0196
.38	6.6184	6.0900	5.6278	5.2215	4.8627
.40	6.4049	5.8935	5.4462	5.0530	4.7059
.42	6.1914	5.6971	5.2647	4.8846	4.5490
.44	5.9779	5.5006	5.0831	4.7162	4.3922
.46	5.7644	5.3042	4.9016	4.5477	4.2353
.48	5.5509	5.1077	4.7201	4.3793	4.0784
.50	5.3374	4.9113	4.5385	4.2109	3.9216
.52	5.1239	4.7148	4.3570	4.0424	3.7647
.54	4.9104	4.5184	4.1754	3.8740	3.6078
.56	4.6969	4.3219	3.9939	3.7056	3.4510
.58	4.4834	4.1255	3.8124	3.5371	3.2941
.60	4.2699	3.9290	3.6308	3.3687	3.1373
.62	4.0564	3.7326	3.4493	3.2003	2.9804
.64	3.8429	3.5361	3.2677	3.0318	2.8235
.66	3.6294	3.3397	3.0862	2.8634	2.6667
.68	3.4159	3.1432	2.9047	2.6950	2.5098
.70	3.2024	2.9468	2.7231	2.5265	2.3529
.72	2.9889	2.7503	2.5416	2.3581	2.1961
.74	2.7754	2.5539	2.3600	2.1897	2.0392
.76	2.5619	2.3574	2.1785	2.0212	1.8824
.78	2.3485	2.1610	1.9969	1.8528	1.7255
.80	2.1350	1.9645	1.8154	1.6843	1.5686
.82	1.9215	1.7681	1.6339	1.5159	1.4118
.84	1.7080	1.5716	1.4523	1.3475	1.2549
.86	1.4945	1.3752	1.2708	1.1790	1.0980
.88	1.2810	1.1787	1.0892	1.0106	.9412
.90	1.0675	.9823	.9077	.8422	.7843

Table 8-15. Fuel Cost Factors

ANNUAL RATE OF FUEL INCREASE = .02
NUMBER OF YEARS = 25

F \ I	0.08	0.09	0.10	0.11	0.12
.10	11.4066	10.4110	9.5465	8.7924	8.1314
.12	11.1532	10.1796	9.3344	8.5970	7.9507
.14	10.8997	9.9483	9.1222	8.4016	7.7700
.16	10.6462	9.7169	8.9101	8.2062	7.5894
.18	10.3927	9.4856	8.6979	8.0108	7.4087
.20	10.1392	9.2542	8.4858	7.8155	7.2280
.22	9.8857	9.0228	8.2736	7.6201	7.0473
.24	9.6323	8.7915	8.0615	7.4247	6.8666
.26	9.3788	8.5601	7.8494	7.2293	6.6859
.28	9.1253	8.3288	7.6372	7.0339	6.5052
.30	8.8718	8.0974	7.4251	6.8385	6.3245
.32	8.6183	7.8661	7.2129	6.6431	6.1438
.34	8.3649	7.6347	7.0008	6.4477	5.9631
.36	8.1114	7.4034	6.7886	6.2524	5.7824
.38	7.8579	7.1720	6.5765	6.0570	5.6017
.40	7.6044	6.9406	6.3643	5.8616	5.4210
.42	7.3509	6.7093	6.1522	5.6662	5.2403
.44	7.0975	6.4779	5.9401	5.4708	5.0596
.46	6.8440	6.2466	5.7279	5.2754	4.8789
.48	6.5905	6.0152	5.5158	5.0800	4.6982
.50	6.3370	5.7839	5.3036	4.8847	4.5175
.52	6.0835	5.5525	5.0915	4.6893	4.3368
.54	5.8301	5.3212	4.8793	4.4939	4.1561
.56	5.5766	5.0898	4.6672	4.2985	3.9754
.58	5.3231	4.8585	4.4550	4.1031	3.7947
.60	5.0696	4.6271	4.2429	3.9077	3.6140
.62	4.8161	4.3957	4.0307	3.7123	3.4333
.64	4.5627	4.1644	3.8186	3.5170	3.2526
.66	4.3092	3.9330	3.6065	3.3216	3.0719
.68	4.0557	3.7017	3.3943	3.1262	2.8912
.70	3.8022	3.4703	3.1822	2.9308	2.7105
.72	3.5487	3.2390	2.9700	2.7354	2.5298
.74	3.2952	3.0076	2.7579	2.5400	2.3491
.76	3.0418	2.7763	2.5457	2.3446	2.1684
.78	2.7883	2.5449	2.3336	2.1492	1.9877
.80	2.5348	2.3135	2.1214	1.9539	1.8070
.82	2.2813	2.0822	1.9093	1.7585	1.6263
.84	2.0278	1.8508	1.6972	1.5631	1.4456
.86	1.7744	1.6195	1.4850	1.3677	1.2649
.88	1.5209	1.3881	1.2729	1.1723	1.0842
.90	1.2674	1.1568	1.0607	.9769	.9035

Table 8-16. Fuel Cost Factors

ANNUAL RATE OF FUEL INCREASE = .04
NUMBER OF YEARS = 25

I F	0.08	0.09	0.10	0.11	0.12
.10	13.7417	12.4353	11.3093	10.3342	9.4859
.12	13.4363	12.1589	11.0580	10.1046	9.2751
.14	13.1309	11.8826	10.8067	9.8749	9.0643
.16	12.8255	11.6063	10.5554	9.6453	8.8535
.18	12.5202	11.3299	10.3040	9.4156	8.6427
.20	12.2148	11.0536	10.0527	9.1860	8.4319
.22	11.9094	10.7772	9.8014	8.9563	8.2211
.24	11.6041	10.5009	9.5501	8.7267	8.0103
.26	11.2987	10.2246	9.2988	8.4970	7.7995
.28	10.9933	9.9482	9.0474	8.2674	7.5887
.30	10.6880	9.6719	8.7961	8.0377	7.3779
.32	10.3826	9.3955	8.5448	7.8081	7.1671
.34	10.0772	9.1192	8.2935	7.5784	6.9563
.36	9.7718	8.8429	8.0422	7.3488	6.7455
.38	9.4665	8.5665	7.7909	7.1191	6.5347
.40	9.1611	8.2902	7.5395	6.8895	6.3239
.42	8.8557	8.0138	7.2882	6.6598	6.1131
.44	8.5504	7.7375	7.0369	6.4302	5.9023
.46	8.2450	7.4612	6.7856	6.2005	5.6915
.48	7.9396	7.1848	6.5343	5.9709	5.4807
.50	7.6343	6.9085	6.2829	5.7412	5.2699
.52	7.3289	6.6321	6.0316	5.5116	5.0591
.54	7.0235	6.3558	5.7803	5.2819	4.8483
.56	6.7181	6.0795	5.5290	5.0523	4.6375
.58	6.4128	5.8031	5.2777	4.8226	4.4267
.60	6.1074	5.5268	5.0264	4.5930	4.2159
.62	5.8020	5.2505	4.7750	4.3633	4.0051
.64	5.4967	4.9741	4.5237	4.1337	3.7943
.66	5.1913	4.6978	4.2724	3.9040	3.5835
.68	4.8859	4.4214	4.0211	3.6744	3.3727
.70	4.5806	4.1451	3.7698	3.4447	3.1620
.72	4.2752	3.8688	3.5185	3.2151	2.9512
.74	3.9698	3.5924	3.2671	2.9854	2.7404
.76	3.6644	3.3161	3.0158	2.7558	2.5296
.78	3.3591	3.0397	2.7645	2.5261	2.3188
.80	3.0537	2.7634	2.5132	2.2965	2.1080
.82	2.7483	2.4871	2.2619	2.0668	1.8972
.84	2.4430	2.2107	2.0105	1.8372	1.6864
.86	2.1376	1.9344	1.7592	1.6075	1.4756
.88	1.8322	1.6580	1.5079	1.3779	1.2648
.90	1.5269	1.3817	1.2566	1.1482	1.0540

Table 8-17. Fuel Cost Factors

ANNUAL RATE OF FUEL INCREASE = .06
NUMBER OF YEARS = 25

I F	0.08	0.09	0.10	0.11	0.12
.10	16.7990	15.0684	13.5872	12.3135	11.2131
.12	16.4256	14.7336	13.2853	12.0399	10.9639
.14	16.0523	14.3987	12.9834	11.7662	10.7147
.16	15.6790	14.0639	12.6814	11.4926	10.4655
.18	15.3057	13.7290	12.3795	11.2190	10.2164
.20	14.9324	13.3942	12.0776	10.9453	9.9672
.22	14.5591	13.0593	11.7756	10.6717	9.7180
.24	14.1858	12.7245	11.4737	10.3981	9.4688
.26	13.8125	12.3896	11.1717	10.1244	9.2196
.28	13.4392	12.0547	10.8698	9.8508	8.9705
.30	13.0659	11.7199	10.5679	9.5772	8.7213
.32	12.6925	11.3850	10.2659	9.3035	8.4721
.34	12.3192	11.0502	9.9640	9.0299	8.2229
.36	11.9459	10.7153	9.6620	8.7563	7.9737
.38	11.5726	10.3805	9.3601	8.4826	7.7246
.40	11.1993	10.0456	9.0582	8.2090	7.4754
.42	10.8260	9.7108	8.7562	7.9354	7.2262
.44	10.4527	9.3759	8.4543	7.6617	6.9770
.46	10.0794	9.0411	8.1523	7.3881	6.7278
.48	9.7061	8.7062	7.8504	7.1145	6.4787
.50	9.3328	8.3714	7.5485	6.8408	6.2295
.52	8.9594	8.0365	7.2465	6.5672	5.9803
.54	8.5861	7.7016	6.9446	6.2936	5.7311
.56	8.2128	7.3668	6.6427	6.0199	5.4819
.58	7.8395	7.0319	6.3407	5.7463	5.2328
.60	7.4662	6.6971	6.0388	5.4727	4.9836
.62	7.0929	6.3622	5.7368	5.1990	4.7344
.64	6.7196	6.0274	5.4349	4.9254	4.4852
.66	6.3463	5.6925	5.1330	4.6518	4.2360
.68	5.9730	5.3577	4.8310	4.3781	3.9869
.70	5.5997	5.0228	4.5291	4.1045	3.7377
.72	5.2263	4.6880	4.2271	3.8309	3.4885
.74	4.8530	4.3531	3.9252	3.5572	3.2393
.76	4.4797	4.0182	3.6233	3.2836	2.9902
.78	4.1064	3.6834	3.3213	3.0100	2.7410
.80	3.7331	3.3485	3.0194	2.7363	2.4918
.82	3.3598	3.0137	2.7174	2.4627	2.2426
.84	2.9865	2.6788	2.4155	2.1891	1.9934
.86	2.6132	2.3440	2.1136	1.9154	1.7443
.88	2.2399	2.0091	1.8116	1.6418	1.4951
.90	1.8666	1.6743	1.5097	1.3682	1.2459

Table 8-18. Fuel Cost Factors

ANNUAL RATE OF FUEL INCREASE = .08
NUMBER OF YEARS = 25

F \ I	0.08	0.09	0.10	0.11	0.12
.10	20.8333	18.5217	16.5561	14.8769	13.4359
.12	20.3704	18.1101	16.1882	14.5463	13.1373
.14	19.9074	17.6986	15.8203	14.2157	12.8387
.16	19.4444	17.2870	15.4524	13.8851	12.5402
.18	18.9815	16.8754	15.0844	13.5545	12.2416
.20	18.5185	16.4638	14.7165	13.2239	11.9430
.22	18.0556	16.0522	14.3486	12.8933	11.6444
.24	17.5926	15.6406	13.9807	12.5627	11.3459
.26	17.1296	15.2290	13.6128	12.2321	11.0473
.28	16.6667	14.8174	13.2449	11.9015	10.7487
.30	16.2037	14.4058	12.8770	11.5709	10.4501
.32	15.7407	13.9942	12.5090	11.2403	10.1516
.34	15.2778	13.5826	12.1411	10.9097	9.8530
.36	14.8148	13.1710	11.7732	10.5791	9.5544
.38	14.3519	12.7594	11.4053	10.2485	9.2558
.40	13.8889	12.3478	11.0374	9.9179	8.9573
.42	13.4259	11.9362	10.6695	9.5873	8.6587
.44	12.9630	11.5246	10.3016	9.2567	8.3601
.46	12.5000	11.1130	9.9337	8.9261	8.0615
.48	12.0370	10.7015	9.5657	8.5955	7.7630
.50	11.5741	10.2899	9.1978	8.2649	7.4644
.52	11.1111	9.8783	8.8299	7.9343	7.1658
.54	10.6481	9.4667	8.4620	7.6038	6.8672
.56	10.1852	9.0551	8.0941	7.2732	6.5687
.58	9.7222	8.6435	7.7262	6.9426	6.2701
.60	9.2593	8.2319	7.3583	6.6120	5.9715
.62	8.7963	7.8203	6.9903	6.2814	5.6729
.64	8.3333	7.4087	6.6224	5.9508	5.3744
.66	7.8704	6.9971	6.2545	5.6202	5.0758
.68	7.4074	6.5855	5.8866	5.2896	4.7772
.70	6.9444	6.1739	5.5187	4.9590	4.4786
.72	6.4815	5.7623	5.1508	4.6284	4.1801
.74	6.0185	5.3507	4.7829	4.2978	3.8815
.76	5.5556	4.9391	4.4150	3.9672	3.5829
.78	5.0926	4.5275	4.0470	3.6366	3.2843
.80	4.6296	4.1159	3.6791	3.3060	2.9858
.82	4.1667	3.7043	3.3112	2.9754	2.6872
.84	3.7037	3.2928	2.9433	2.6448	2.3886
.86	3.2407	2.8812	2.5754	2.3142	2.0900
.88	2.7778	2.4696	2.2075	1.9836	1.7915
.90	2.3148	2.0580	1.8396	1.6530	1.4929

Table 8-19. Fuel Cost Factors

ANNUAL RATE OF FUEL INCREASE = .10
NUMBER OF YEARS = 25

F. \ I	0.08	0.09	0.10	0.11	0.12
.10	26.1927	23.0830	20.4545	18.2230	16.3200
.12	25.6107	22.5700	20.0000	17.8181	15.9573
.14	25.0286	22.0571	19.5455	17.4131	15.5947
.16	24.4466	21.5441	19.0909	17.0082	15.2320
.18	23.8645	21.0311	18.6364	16.6032	14.8693
.20	23.2824	20.5182	18.1818	16.1982	14.5067
.22	22.7004	20.0052	17.7273	15.7933	14.1440
.24	22.1183	19.4923	17.2727	15.3883	13.7813
.26	21.5363	18.9793	16.8182	14.9834	13.4187
.28	20.9542	18.4664	16.3636	14.5784	13.0560
.30	20.3721	17.9534	15.9091	14.1735	12.6933
.32	19.7901	17.4405	15.4545	13.7685	12.3307
.34	19.2080	16.9275	15.0000	13.3635	11.9680
.36	18.6260	16.4146	14.5455	12.9586	11.6053
.38	18.0439	15.9016	14.0909	12.5536	11.2427
.40	17.4618	15.3886	13.6364	12.1487	10.8800
.42	16.8798	14.8757	13.1818	11.7437	10.5173
.44	16.2977	14.3627	12.7273	11.3388	10.1547
.46	15.7156	13.8498	12.2727	10.9338	9.7920
.48	15.1336	13.3368	11.8182	10.5289	9.4293
.50	14.5515	12.8239	11.3636	10.1239	9.0667
.52	13.9695	12.3109	10.9091	9.7189	8.7040
.54	13.3874	11.7980	10.4545	9.3140	8.3413
.56	12.8053	11.2850	10.0000	8.9090	7.9787
.58	12.2233	10.7721	9.5455	8.5041	7.6160
.60	11.6412	10.2591	9.0909	8.0991	7.2533
.62	11.0592	9.7461	8.6364	7.6942	6.8907
.64	10.4771	9.2332	8.1818	7.2892	6.5280
.66	9.8950	8.7202	7.7273	6.8843	6.1653
.68	9.3130	8.2073	7.2727	6.4793	5.8027
.70	8.7309	7.6943	6.8182	6.0743	5.4400
.72	8.1489	7.1814	6.3636	5.6694	5.0773
.74	7.5668	6.6684	5.9091	5.2644	4.7147
.76	6.9847	6.1555	5.4545	4.8595	4.3520
.78	6.4027	5.6425	5.0000	4.4545	3.9893
.80	5.8206	5.1295	4.5455	4.0496	3.6267
.82	5.2385	4.6166	4.0909	3.6446	3.2640
.84	4.6565	4.1036	3.6364	3.2396	2.9013
.86	4.0744	3.5907	3.1818	2.8347	2.5387
.88	3.4924	3.0777	2.7273	2.4297	2.1760
.90	2.9103	2.5648	2.2727	2.0248	1.8133

Table 8-20. Fuel Cost Factors

ANNUAL RATE OF FUEL INCREASE = .12
NUMBER OF YEARS = 25

F \ I	0.08	0.09	0.10	0.11	0.12
.10	33.3521	29.1438	25.6067	22.6208	20.0893
.12	32.6109	28.4962	25.0377	22.1181	19.6429
.14	31.8697	27.8485	24.4686	21.6154	19.1964
.16	31.1286	27.2009	23.8996	21.1127	18.7500
.18	30.3874	26.5533	23.3306	20.6101	18.3036
.20	29.6463	25.9056	22.7615	20.1074	17.8571
.22	28.9051	25.2580	22.1925	19.6047	17.4107
.24	28.1640	24.6103	21.6234	19.1020	16.9643
.26	27.4228	23.9627	21.0544	18.5993	16.5179
.28	26.6816	23.3151	20.4854	18.0966	16.0714
.30	25.9405	22.6674	19.9163	17.5940	15.6250
.32	25.1993	22.0198	19.3473	17.0913	15.1786
.34	24.4582	21.3721	18.7783	16.5886	14.7321
.36	23.7170	20.7245	18.2093	16.0859	14.2857
.38	22.9759	20.0769	17.6402	15.5832	13.8393
.40	22.2347	19.4292	17.0711	15.0805	13.3929
.42	21.4935	18.7816	16.5021	14.5778	12.9464
.44	20.7524	18.1339	15.9331	14.0752	12.5000
.46	20.0112	17.4863	15.3640	13.5725	12.0536
.48	19.2701	16.8387	14.7950	13.0698	11.6071
.50	18.5289	16.1910	14.2259	12.5671	11.1607
.52	17.7878	15.5434	13.6569	12.0644	10.7143
.54	17.0466	14.8957	13.0879	11.5617	10.2679
.56	16.3055	14.2481	12.5188	11.0591	9.8214
.58	15.5643	13.6004	11.9498	10.5564	9.3750
.60	14.8231	12.9528	11.3808	10.0537	8.9286
.62	14.0820	12.3052	10.8117	9.5510	8.4821
.64	13.3408	11.6575	10.2427	9.0483	8.0357
.66	12.5997	11.0099	9.6736	8.5456	7.5893
.68	11.8585	10.3622	9.1046	8.0430	7.1429
.70	11.1174	9.7146	8.5356	7.5403	6.6964
.72	10.3762	9.0670	7.9665	7.0376	6.2500
.74	9.6350	8.4193	7.3975	6.5349	5.8036
.76	8.8939	7.7717	6.8285	6.0322	5.3571
.78	8.1527	7.1240	6.2594	5.5295	4.9107
.80	7.4116	6.4764	5.6904	5.0268	4.4643
.82	6.6704	5.8288	5.1213	4.5242	4.0179
.84	5.9293	5.1811	4.5523	4.0215	3.5714
.86	5.1881	4.5335	3.9833	3.5188	3.1250
.88	4.4469	3.8858	3.4142	3.0161	2.6786
.90	3.7058	3.2382	2.8452	2.5134	2.2321

Table 8-21. Fuel Cost Factors

ANNUAL RATE OF FUEL INCREASE =0.00
NUMBER OF YEARS = 30

I F	0.08	0.09	0.10	0.11	0.12
.10	10.1320	9.2463	8.4842	7.8244	7.2497
.12	9.9068	9.0408	8.2957	7.6505	7.0886
.14	9.6817	8.8353	8.1071	7.4767	6.9275
.16	9.4565	8.6299	7.9186	7.3028	6.7664
.18	9.2314	8.4244	7.7301	7.1289	6.6053
.20	9.0062	8.2189	7.5415	6.9550	6.4441
.22	8.7811	8.0135	7.3530	6.7812	6.2830
.24	8.5559	7.8080	7.1645	6.6073	6.1219
.26	8.3308	7.6025	6.9759	6.4334	5.9608
.28	8.1056	7.3970	6.7874	6.2595	5.7997
.30	7.8804	7.1916	6.5988	6.0857	5.6386
.32	7.6553	6.9861	6.4103	5.9118	5.4775
.34	7.4301	6.7806	6.2218	5.7379	5.3164
.36	7.2050	6.5751	6.0332	5.5640	5.1553
.38	6.9798	6.3697	5.8447	5.3902	4.9942
.40	6.7547	6.1642	5.6561	5.2163	4.8331
.42	6.5295	5.9587	5.4676	5.0424	4.6720
.44	6.3044	5.7532	5.2791	4.8685	4.5109
.46	6.0792	5.5478	5.0905	4.6946	4.3498
.48	5.8540	5.3423	4.9020	4.5208	4.1887
.50	5.6289	5.1368	4.7135	4.3469	4.0276
.52	5.4037	4.9314	4.5249	4.1730	3.8665
.54	5.1786	4.7259	4.3364	3.9991	3.7054
.56	4.9534	4.5204	4.1478	3.8253	3.5443
.58	4.7283	4.3149	3.9593	3.6514	3.3832
.60	4.5031	4.1095	3.7708	3.4775	3.2221
.62	4.2780	3.9040	3.5822	3.3036	3.0610
.64	4.0528	3.6985	3.3937	3.1298	2.8999
.66	3.8276	3.4930	3.2052	2.9559	2.7388
.68	3.6025	3.2876	3.0166	2.7820	2.5777
.70	3.3773	3.0821	2.8281	2.6081	2.4166
.72	3.1522	2.8766	2.6395	2.4343	2.2555
.74	2.9270	2.6712	2.4510	2.2604	2.0943
.76	2.7019	2.4657	2.2625	2.0865	1.9332
.78	2.4767	2.2602	2.0739	1.9126	1.7721
.80	2.2516	2.0547	1.8854	1.7388	1.6110
.82	2.0264	1.8493	1.6968	1.5649	1.4499
.84	1.8012	1.6438	1.5083	1.3910	1.2888
.86	1.5761	1.4383	1.3198	1.2171	1.1277
.88	1.3509	1.2328	1.1312	1.0433	.9666
.90	1.1258	1.0274	.9427	.8694	.8055

Table 8-22. Fuel Cost Factors

ANNUAL RATE OF FUEL INCREASE = .02
NUMBER OF YEARS = 30

I F	0.08	0.09	0.10	0.11	0.12
.10	12.2999	11.1018	10.0822	9.2087	8.4559
.12	12.0265	10.8551	9.8581	9.0041	8.2680
.14	11.7532	10.6084	9.6341	8.7995	8.0800
.16	11.4799	10.3617	9.4100	8.5948	7.8921
.18	11.2066	10.1150	9.1860	8.3902	7.7042
.20	10.9332	9.8683	8.9619	8.1856	7.5163
.22	10.6599	9.6216	8.7379	7.9309	7.3284
.24	10.3866	9.3749	8.5138	7.7763	7.1405
.26	10.1132	9.1282	8.2898	7.5716	6.9526
.28	9.8399	8.8815	8.0657	7.3670	6.7647
.30	9.5666	8.6348	7.8417	7.1624	6.5768
.32	9.2932	8.3880	7.6176	6.9577	6.3889
.34	9.0199	8.1413	7.3936	6.7531	6.2010
.36	8.7466	7.8946	7.1695	6.5484	6.0131
.38	8.4732	7.6479	6.9455	6.3438	5.8252
.40	8.1999	7.4012	6.7215	6.1392	5.6372
.42	7.9266	7.1545	6.4974	5.9345	5.4493
.44	7.6533	6.9078	6.2734	5.7299	5.2614
.46	7.3799	6.6611	6.0493	5.5252	5.0735
.48	7.1066	6.4144	5.8253	5.3206	4.8856
.50	6.8333	6.1677	5.6012	5.1160	4.6977
.52	6.5599	5.9210	5.3772	4.9113	4.5098
.54	6.2866	5.6743	5.1531	4.7067	4.3219
.56	6.0133	5.4276	4.9291	4.5021	4.1340
.58	5.7399	5.1809	4.7050	4.2974	3.9461
.60	5.4666	4.9341	4.4810	4.0928	3.7582
.62	5.1933	4.6874	4.2569	3.8881	3.5703
.64	4.9200	4.4407	4.0329	3.6835	3.3823
.66	4.6466	4.1940	3.8088	3.4789	3.1944
.68	4.3733	3.9473	3.5848	3.2742	3.0065
.70	4.1000	3.7006	3.3607	3.0696	2.8186
.72	3.8266	3.4539	3.1367	2.8649	2.6307
.74	3.5533	3.2072	2.9126	2.6603	2.4428
.76	3.2800	2.9605	2.6886	2.4557	2.2549
.78	3.0066	2.7138	2.4645	2.2510	2.0670
.80	2.7333	2.4671	2.2405	2.0464	1.8791
.82	2.4600	2.2204	2.0164	1.8417	1.6912
.84	2.1866	1.9737	1.7924	1.6371	1.5033
.86	1.9133	1.7270	1.5683	1.4325	1.3154
.88	1.6400	1.4802	1.3443	1.2278	1.1274
.90	1.3667	1.2335	1.1202	1.0232	.9395

Table 8-23. Fuel Cost Factors

ANNUAL RATE OF FUEL INCREASE = .04
NUMBER OF YEARS = 30

F \ I	0.08	0.09	0.10	0.11	0.12
.10	15.2478	13.5997	12.2119	11.0355	10.0321
.12	14.9090	13.2975	11.9405	10.7903	9.8092
.14	14.5701	12.9953	11.6691	10.5451	9.5862
.16	14.2313	12.6931	11.3978	10.2998	9.3633
.18	13.8924	12.3909	11.1264	10.0546	9.1404
.20	13.5536	12.0887	10.8550	9.8094	8.9174
.22	13.2148	11.7864	10.5836	9.5641	8.6945
.24	12.8759	11.4842	10.3123	9.3189	8.4716
.26	12.5371	11.1820	10.0409	9.0737	8.2486
.28	12.1982	10.8798	9.7695	8.8284	8.0257
.30	11.8594	10.5776	9.4981	8.5832	7.8027
.32	11.5206	10.2754	9.2268	8.3380	7.5798
.34	11.1817	9.9731	8.9554	8.0927	7.3569
.36	10.8429	9.6709	8.6840	7.8475	7.133[illegible]
.38	10.5040	9.3687	8.4126	7.6023	6.9110
.40	10.1652	9.0665	8.1413	7.3570	6.6881
.42	9.8264	8.7643	7.8699	7.1118	6.4651
.44	9.4875	8.4621	7.5985	6.8666	6.2422
.46	9.1487	8.1598	7.3271	6.6213	6.0193
.48	8.8098	7.8576	7.0558	6.3761	5.7963
.50	8.4710	7.5554	6.7844	6.1309	5.5734
.52	8.1322	7.2532	6.5130	5.8856	5.3505
.54	7.7933	6.9510	6.2416	5.6404	5.1275
.56	7.4545	6.6488	5.9703	5.3951	4.9046
.58	7.1156	6.3465	5.6989	5.1499	4.6816
.60	6.7768	6.0443	5.4275	4.9047	4.4587
.62	6.4380	5.7421	5.1561	4.6594	4.2358
.64	6.0991	5.4399	4.8848	4.4142	4.0128
.66	5.7603	5.1377	4.6134	4.1690	3.7899
.68	5.4214	4.8355	4.3420	3.9237	3.5670
.70	5.0826	4.5332	4.0706	3.6785	3.3440
.72	4.7438	4.2310	3.7993	3.4333	3.1211
.74	4.4049	3.9288	3.5279	3.1880	2.8982
.76	4.0661	3.6266	3.2565	2.9428	2.6752
.78	3.7272	3.3244	2.9851	2.6976	2.4523
.80	3.3884	3.0222	2.7138	2.4523	2.2294
.82	3.0496	2.7199	2.4424	2.2071	2.0064
.84	2.7107	2.4177	2.1710	1.9619	1.7835
.86	2.3719	2.1155	1.8996	1.7166	1.5605
.88	2.0330	1.8133	1.6283	1.4714	1.3376
.90	1.6942	1.5111	1.3569	1.2262	1.1147

Table 8-24. Fuel Cost Factors

ANNUAL RATE OF FUEL INCREASE = .06
NUMBER OF YEARS = 30

F \ I	0.08	0.09	0.10	0.11	0.12
.10	19.3152	17.0132	15.0941	13.4839	12.1244
.12	18.8860	16.6351	14.7587	13.1843	11.8550
.14	18.4568	16.2571	14.4232	12.8847	11.5856
.16	18.0275	15.8790	14.0878	12.5850	11.3161
.18	17.5983	15.5009	13.7524	12.2854	11.0467
.20	17.1691	15.1228	13.4170	11.9857	10.7773
.22	16.7399	14.7448	13.0816	11.6861	10.5078
.24	16.3106	14.3667	12.7461	11.3864	10.2384
.26	15.8814	13.9886	12.4107	11.0868	9.9690
.28	15.4522	13.6106	12.0753	10.7872	9.6995
.30	15.0230	13.2325	11.7399	10.4875	9.4301
.32	14.5937	12.8544	11.4044	10.1879	9.1607
.34	14.1645	12.4763	11.0690	9.8882	8.8912
.36	13.7353	12.0983	10.7336	9.5886	8.6218
.38	13.3060	11.7202	10.3982	9.2889	8.3524
.40	12.8768	11.3421	10.0627	8.9893	8.0829
.42	12.4476	10.9641	9.7273	8.6897	7.8135
.44	12.0184	10.5860	9.3919	8.3900	7.5441
.46	11.5891	10.2079	9.0565	8.0904	7.2746
.48	11.1599	9.8298	8.7210	7.7907	7.0052
.50	10.7307	9.4518	8.3856	7.4911	6.7358
.52	10.3015	9.0[illegible]	8.0502	7.1914	6.4664
.54	9.8722	8.6956	7.7148	6.8918	6.1969
.56	9.4430	8.3176	7.3793	6.5922	5.9275
.58	9.0138	7.9395	7.0439	6.2925	5.6581
.60	8.5845	7.5614	6.7085	5.9929	5.3886
.62	8.1553	7.1833	6.3731	5.6932	5.1192
.64	7.7261	6.8053	6.0376	5.3936	4.8498
.66	7.2969	6.4272	5.7022	5.0939	4.5803
.68	6.8676	6.0491	5.3668	4.7943	4.3109
.70	6.4384	5.6711	5.0314	4.4946	4.0415
.72	6.0092	5.2930	4.6959	4.1950	3.7720
.74	5.5800	4.9149	4.3605	3.8954	3.5026
.76	5.1507	4.5369	4.0251	3.5957	3.2332
.78	4.7215	4.1588	3.6897	3.2961	2.9637
.80	4.2923	3.7807	3.3542	2.9964	2.6943
.82	3.8630	3.4026	3.0188	2.6968	2.4249
.84	3.4338	3.0246	2.6834	2.3971	2.1555
.86	3.0046	2.6465	2.3480	2.0975	1.8860
.88	2.5754	2.2684	2.0125	1.7979	1.6166
.90	2.1461	1.8904	1.6771	1.4982	1.3472

Table 8-25. Fuel Cost Factors

ANNUAL RATE OF FUEL INCREASE = .08
NUMBER OF YEARS = 30

I F	0.08	0.09	0.10	0.11	0.12
.10	25.0000	21.7410	19.0496	16.8130	14.9429
.12	24.4444	21.2578	18.6262	16.4394	14.6108
.14	23.8889	20.7747	18.2029	16.0658	14.2788
.16	23.3333	20.2916	17.7796	15.6922	13.9467
.18	22.7778	19.8084	17.3563	15.3186	13.6147
.20	22.2222	19.3253	16.9329	14.9449	13.2826
.22	21.6667	18.8422	16.5096	14.5713	12.9505
.24	21.1111	[illegible]590	16.0863	14.1977	12.6185
.26	20.5556	1[illegible]759	15.6630	13.8241	12.2864
.28	20.0000	17[illegible]928	15.2397	13.45[illegible]4	11.9543
.30	19.4444	16.9096	14.8163	13.0768	11.6223
.32	18.8889	16.4265	14.3930	12.7032	11.2902
.34	18.3333	15.9434	13.9697	12.3296	10.9581
.36	17.7778	15.4602	13.5464	11.9559	10.6261
.38	17.2222	14.9771	13.1230	11.5823	10.2940
.40	16.6667	14.4940	12.6997	11.2087	9.9619
.42	16.1111	14.0108	12.2764	10.8351	9.6299
.44	15.5556	13.5277	11.8531	10.4614	9.2978
.46	15.0000	13.0446	11.4297	10.0878	8.9657
.48	14.4444	12.5614	11.0064	9.7142	8.6337
.50	13.8889	12.0783	10.5831	9.3406	8.3016
.52	13.3333	11.5952	10.1598	8.9670	7.9696
.54	12.7778	11.1120	9.7364	8.5933	7.6375
.56	12.2222	10.6289	9.3131	8.2197	7.3054
.58	11.6667	10.1458	8.8898	7.8461	6.9734
.60	11.1111	9.6626	8.4665	7.4725	6.6413
.62	10.5556	9.1795	8.0432	7.0988	6.3092
.64	10.0000	8.6964	7.6198	6.7252	5.9772
.66	9.4444	8.2132	7.1965	6.3516	5.6451
.68	8.8889	7.7301	6.7732	5.9780	5.3130
.70	8.3333	7.2470	6.3499	5.6043	4.9810
.72	7.7778	6.7639	5.9265	5.2307	4.6489
.74	7.2222	6.2807	5.5032	4.8571	4.3159
.76	6.6667	5.7976	5.0799	4.4835	3.9849
.78	6.1111	5.3145	4.6566	4.1099	3.6527
.80	5.5556	4.8313	4.2332	3.7362	3.3206
.82	5.0000	4.3482	3.8099	3.3626	2.9886
.84	4.4444	3.8651	3.3866	2.9890	2.6565
.86	3.8889	3.3819	2.9633	2.6154	2.3245
.88	3.3333	2.8988	2.5399	2.2417	1.9924
.90	2.7778	2.4157	2.1166	1.8681	1.6603

Table 8-26. Fuel Cost Factors

ANNUAL RATE OF FUEL INCREASE = .10
NUMBER OF YEARS = 30

F \ I	0.08	0.09	0.10	0.11	0.12
.10	33.0334	28.3663	24.5455	21.3985	18.7909
.12	32.2993	27.7360	24.0000	20.9230	18.3733
.14	31.5652	27.1056	23.4545	20.4474	17.9557
.16	30.8312	26.4752	22.9091	19.9719	17.5382
.18	30.0971	25.8449	22.3636	19.4964	17.1206
.20	29.3630	25.2145	21.8182	19.0209	16.7030
.22	28.6289	24.5841	21.2727	18.5454	16.2854
.24	27.8949	23.9538	20.7273	18.0698	15.8679
.26	27.1608	23.3234	20.1818	17.5943	15.4503
.28	26.4267	22.6931	19.6364	17.1188	15.0327
.30	25.6926	22.0627	19.0909	16.6433	14.6151
.32	24.9586	21.4323	18.5455	16.1677	14.1976
.34	24.2245	20.8020	18.0000	15.6922	13.7800
.36	23.4904	20.1716	17.4545	15.2167	13.3624
.38	22.7563	19.5412	16.9091	14.7412	12.9448
.40	22.0223	18.9109	16.3636	14.2657	12.5273
.42	21.2882	18.2805	15.8182	13.7901	12.1097
.44	20.5541	17.6502	15.2727	13.3146	11.6921
.46	19.8200	17.0198	14.7273	12.8391	11.2745
.48	19.0860	16.3894	14.1818	12.3636	10.8570
.50	18.3519	15.7591	13.6364	11.8880	10.4394
.52	17.6178	15.1287	13.0909	11.4125	10.0218
.54	16.8837	14.4983	12.5455	10.9370	9.6042
.56	16.1497	13.8680	12.0000	10.4615	9.1867
.58	15.4156	13.2376	11.4545	9.9860	8.7691
.60	14.6815	12.6073	10.9091	9.5104	8.3515
.62	13.9474	11.9769	10.3636	9.0349	7.9339
.64	13.2134	11.3465	9.8182	8.5594	7.5164
.66	12.4793	10.7162	9.2727	8.0839	7.0988
.68	11.7452	10.0858	8.7273	7.6084	6.6812
.70	11.0111	9.4554	8.1818	7.1328	6.2636
.72	10.2771	8.8251	7.6364	6.6573	5.8461
.74	9.5430	8.1947	7.0909	6.1818	5.4285
.76	8.8089	7.5644	6.5455	5.7063	5.0109
.78	8.0748	6.9340	6.0000	5.2307	4.5933
.80	7.3408	6.3036	5.4545	4.7552	4.1758
.82	6.6067	5.6733	4.9091	4.2797	3.7582
.84	5.8726	5.0429	4.3636	3.8042	3.3406
.86	5.1385	4.4125	3.8182	3.3287	2.9230
.88	4.4045	3.7822	3.2727	2.8531	2.5055
.90	3.6704	3.1518	2.7273	2.3776	2.0879

Table 8-27. Fuel Cost Factors

ANNUAL RATE OF FUEL INCREASE = .12
NUMBER OF YEARS = 30

F \ I	0.08	0.09	0.10	0.11	0.12
.10	44.4901	37.7434	32.2632	27.7860	24.1071
.12	43.5014	36.9047	31.5462	27.1586	23.5714
.14	42.5127	36.0659	30.8293	26.5511	23.0357
.16	41.5241	35.2272	30.1123	25.9336	22.5000
.18	40.5354	34.3884	29.3954	25.3162	21.9643
.20	39.5467	33.5497	28.6784	24.6987	21.4286
.22	38.5581	32.7109	27.9614	24.0812	20.8929
.24	37.5694	31.8722	27.2445	23.4638	20.3571
.26	36.5807	31.0335	26.5275	22.8463	19.8214
.28	35.5921	30.1947	25.8106	22.2288	19.2857
.30	34.6034	29.3560	25.0936	21.6114	18.7500
.32	33.6147	28.5172	24.3766	20.9939	18.2143
.34	32.6261	27.6785	23.6597	20.3764	17.6786
.36	31.6374	26.9398	22.9427	19.7590	17.1429
.38	30.6487	26.0010	22.2258	19.1415	16.6071
.40	29.6601	25.1623	21.5088	18.5240	16.0714
.42	28.6714	24.3235	20.7918	17.9066	15.5357
.44	27.6827	23.4848	20.0749	17.2891	15.0000
.46	26.6941	22.6460	19.3579	16.6716	14.4643
.48	25.7054	21.8073	18.6410	16.0542	13.9286
.50	24.7167	20.9686	17.9240	15.4367	13.3929
.52	23.7280	20.1298	17.2070	14.8192	12.8571
.54	22.7394	19.2911	16.4901	14.2018	12.3214
.56	21.7507	18.4523	15.7731	13.5843	11.7857
.58	20.7620	17.6136	15.0562	12.9668	11.2500
.60	19.7734	16.7748	14.3392	12.3494	10.7143
.62	18.7847	15.9361	13.6222	11.7319	10.1786
.64	17.7960	15.0974	12.9053	11.1144	9.6429
.66	16.8074	14.2586	12.1883	10.4969	9.1071
.68	15.8187	13.4199	11.4714	9.8795	8.5714
.70	14.8300	12.5811	10.7544	9.2620	8.0357
.72	13.8414	11.7424	10.0374	8.6445	7.5000
.74	12.8527	10.9036	9.3205	8.0271	6.9643
.76	11.8640	10.0649	8.6035	7.4096	6.4286
.78	10.8754	9.2262	7.8866	6.7921	5.8929
.80	9.8867	8.3874	7.1696	6.1747	5.3571
.82	8.8980	7.5487	6.4526	5.5572	4.8214
.84	7.9093	6.7099	5.7357	4.9397	4.2857
.86	6.9207	5.8712	5.0187	4.3223	3.7500
.88	5.9320	5.0325	4.3018	3.7048	3.2143
.90	4.9433	4.1937	3.5848	3.0873	2.6786

TRAINING COURSE IN
THE PRACTICAL ASPECTS OF
DESIGN OF SOLAR HEATING AND COOLING SYSTEMS
FOR
RESIDENTIAL BUILDINGS

MODULE 9

ENERGY CONSERVATION TRADE-OFFS

SOLAR ENERGY APPLICATIONS LABORATORY
COLORADO STATE UNIVERSITY
FORT COLLINS, COLORADO

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INTRODUCTION

TRAINEE-ORIENTED OBJECTIVE: To understand energy-conservation methods and be able to choose and incorporate the most cost-effective methods of saving energy

SUB-OBJECTIVES: At the end of the module the trainee should be able to:

1. Obtain an overview of projected cost and implications
2. Identify and describe three major factors in excess energy usage of
    a. Building design
    b. Building operation
    c. Building maintenance
3. Describe the energy saving benefits of
    a. Glass area and type
    b. External shading
    c. Proper sizing of heating/cooling equipment
    d. Buffering of entryways
    e. Insulation
4. Design cost-performance models considering material, labor, and overhead
5. Describe the benefits of using load models to determine cost savings
6. Utilize load models to determine cost savings
7. Describe the sensitivity to change in costs of energy, construction, maintenance, operation, and investment funds.

It is not the intent in this module to discuss all potential energy-conservation measures, but rather to point out a few and present a few detailed examples where the measures are significantly beneficial, especially for solar heating and/or cooling systems.

In the past, home design has been based on an abundant supply of cheap energy. Now that these conditions of supply and cost are changing, the reconsideration of heating and cooling equipment, architectural designs, construction methods, and home-user practices is necessary. More effective means must be devised to make efficient use of energy in residential buildings. There are approximately 70 million residential living units in the United States and they account for approximately one-fifth of the nation's energy consumption. Of the total energy produced in the United States, approximately 11 percent goes toward residential space heating. A small, but potentially significant amount of the total electric power generated is attributable to cooling of residential buildings. The potential of energy savings that can be realized in building construction makes this subject as important as developing new energy sources.

Federal agencies are concerned with establishing energy-conservation measures and practices in existing as well as new buildings. Guidelines are available through regional offices of these governmental agencies. In addition, conservation information is available through the National Association of Home Builders and public utility companies.

This module is concerned with four areas of energy impact:

1. Architectural emphasis
2. The materials to be used in the construction of the building
3. The proper selection of heating and cooling equipment
4. The home-user practices for operating and maintaining the residential dwelling.

Any energy-conservation modification must be based upon cost-performance (or benefit) decisions.

ENERGY CONSERVATION

The recent concern over increasing prices for fuels that are in short supply has drawn attention to the need to improve the efficiency of house and commercial space heating and cooling. The use of solar heating, and, in some cases, solar cooling, should reduce the cost of these improvements over the whole life of a house in locations where the cost of solar energy is not too high, and fuel costs are not too low. Heating and cooling capital and operating costs presently amount to 10 percent of mortgage costs or nearly 3 percent of the gross earnings of the average U.S. home owner. Since these percentages will likely increase with time, improvements in the thermal design of homes and other buildings may be justified now. In most cases, a decision has been made as to the best value of insulation and other thermal enhancements for the entire life of a building prior to construction. These decisions tend to be irrevocable, in light of the high cost of building modifications or retrofits. Consequently, it is necessary that design data and analysis procedures are used prior to initial building design. In the final analysis, uncertainties as to future fuel prices make all such designs no more than calculated guesses as to what the optimum modifications or changes to consider in the thermal design of a building should be.

As an example, Table 9-1 indicates some general trends for energy conservation. The left-hand column indicates typical geographical characteristics, and the column headings along the top of the page indicate potential energy-conservation measures. Double plusses in the table indicate a strong positive effect of the energy conservation measure for the particular geographical characteristic; single plus signs indicate a moderately positive effect; a zero indicates little, if any, positive effect; and NR indicates a "not relevant" effect. Where the cost of fuel is great, most of the conservation modifications yield strong positive effects. Obviously, where the cost of energy is less, there is less benefit derived. This table is not meant to be all-inclusive, but rather to give some general trends and some energy-conservation measures that the designer should keep in mind for a new building. In addition to the general chart, Tables 9-2 and 9-3 give examples of retrofit packages which have been proposed by the National Association of Home Builders to be used as entire packages for energy-conservative retrofit businesses. Some examples of energy-conservative design changes will be used later in this module to indicate possible cost savings, and whether the cost savings are economical.

There are three major factors contributing to excess energy usage. These factors involve building design, operation, and maintenance.

TABLE 9-1. Energy-Conserving Modifications

Geographical Characteristics	Storm Window	Double Glazed Windows	Storm Doors	Weather Stripped Windows & Doors	Reduced Window Area	Increased Wall Insulation	Reduced Window Area and Storm Windows	Thermostat Night Set-back (Heating)	Heat Recovery Unit for Furnace Flue Gas	Automatic Flue Shut-Off for Furnace	High Performance Air Conditioner	Open Air Cycle for Air Conditioner	Orientation (Long Axis)	Reduction in Internal Load
DRY-BULB TEMPERATURE														
Cold	++	++	++	+	+	++	++	+	+	+	0	0	+	0
Moderate	+	+	+	+	+	+	+	+	+	+	+	+	+	0
Warm	0	0	0	0	0	0	0	0	0	0	++	+	++	++
HUMIDITY*														
High	+	+	+	+	NR	+	+	+	NR	NR	++	0	NR	NR
Low	0	0	0	0	NR	0	0	0	NR	NR	+	+	NR	NR
WIND VELOCITY														
High	+	0	++	++	+	0	++	++	0	+	+	0	Depends on pre-vailing wind direction	NR
Low	0	0	+	+	0	0	+	+	0	0	0	0		NR
SOLAR RADIATION														
High	0	0	0	0	0	+	0	NR	0	0	+	0	++	NR
Low	0	0	0	0	0	0	0	NR	0	0	0	0	+	NR
CONVENTIONAL HEATING FUEL														
Electricity	++	++	++	++	++	++	++	++	NR	NR	NR	NR	+	0
Gas	+	+	+	+	+	+	+	+	+	+	NR	NR	+	0
Oil	+	+	+	+	+	+	+	+	+	+	NR	NR	+	0
ENERGY COSTS														
High	++	++	++	++	++	++	++	++	++	++	++	+	+	+
Low	+	+	+	+	+	+	+	+	+	+	+	0	0	+

++ = Strong positive effect; + = moderately positive effect; 0 = little, if any effect; NR = not relevant.
* = In very low outside temperature, inside humidity limited by condensation on walls and windows.

TABLE 9-2. Retrofit Packages for Zone 1 and Zone 2.

ZONE 1
(Less than 4500 DD)

BASIC PACKAGE: Includes any or all items below, where applicable--

- Install day/night clock thermostat
- Add ceiling insulation to achieve total of approximately R-19
- Tune-up furnace/air conditioning system
- Weatherstrip exterior doors
- Seal all openings and cracks in exterior walls
- Calibrate water heater temperature
- Weatherstrip and insulate attic access door
- Inspection of entire house for additional recommendations

BETTER PACKAGE: Includes items under Basic Package plus the following items, where applicable--

- Add ceiling insulation to achieve total of approximately R-22
- Install R-11 floor insulation over unconditioned spaces
- Tape joints and insulate ducts in unconditioned spaces
- Install storm windows, all living areas
- Blow insulation in uninsulated exterior wall cavities (R-11)
- Add or increase natural attic ventilation, if necessary
- Install ceiling fan for summer cooling

ZONE 2
(4500 - 8000 DD)

BASIC PACKAGE: Includes any or all items below, where applicable--

- Install day/night clock thermostat
- Add ceiling insulation to achieve total of approximately R-22
- Install storm windows, all living areas
- Tune-up furnace/air conditioning system
- Install storm doors and weatherstrip exterior prime doors
- Seal all openings and cracks in exterior walls
- Calibrate water heater temperature
- Weatherstrip and insulate attic access door
- Inspection of entire house for additional recommendations

BETTER PACKAGE: Includes items under Basic Package plus the following items, where applicable--

- Add ceiling insulation to achieve total of approximately R-30
- Install R-11 floor insulation over unconditioned spaces
- Tape joints and insulate ducts in unconditioned spaces
- Blow insulation in uninsulated exterior wall cavities
- Install 2 x 2 furring and R-7 insulation to basement walls
- Install basement storm windows and doors
- Add or increase natural attic ventilation

TABLE 9-3. Retrofit Packages for Zone 3

ZONE 3
(Greater than 8000 DD)

BASIC PACKAGE: Includes any or all items below, where applicable--

- Install day/night clock thermostat
- Add ceiling insulation to achieve total of approximately R-30
- Install storm windows, all living areas
- Tune-up furnace/air conditioning system
- Install storm doors and weatherstrip exterior prime doors
- Seal all openings and cracks in exterior walls
- Calibrate water heater temperature
- Weatherstrip and insulate attic access door
- Inspection of entire house for additional recommendations

BETTER PACKAGE: Includes items under Basic Package plus the following items, where applicable--

- Add ceiling insulation to achieve total of approximately R-38
- Install R-11 floor insulation over unconditioned spaces
- Tape joints and insulate ducts in unconditioned spaces
- Blow insulation in uninsulated exterior wall cavities (R-11)
- Install 2x3 framing (1" from wall) and R-11 insulation to basement walls
- Install basement storm windows and doors
- Add or increase natural attic ventilation

CUSTOM OPTIONS: The following items are in addition to the "BASIC" or "BETTER" packages. They apply to any of the three zones and depend on owner's preference.--

- Replace shower heads with hot water saving type and replace defective washers in all faucets
- Build in vestibule at entrances
- Replace electric resistance heating with heat pump
- Replace furnace heating system with properly sized efficient units
- Replace air conditioning with properly sized, high EER equipment
- Install awnings on East/West windows
- Install attic exhaust fan
- Install exhaust fan in window, wall or ceiling beneath attic for summer cooling
- Install fan(s) hung from ceiling for summer cooling
- Replace incandescent lighting with fluorescent
- Modify roof overhang for summer shading

ERIC
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BUILDING DESIGN

Design in this context also includes construction. In many cases, the thermal characteristics of the building are greatly affected by the construction. Poor workmanship for installing insulation, fitting doors and windows will not provide the expected R values and will increase infiltration losses. The major factors in the building design are material selection and building orientation. For example, glass area is important because of heat losses and gains through windows. Heat losses and gains through windows are larger than through the walls. For this reason it is very important that the ratio of window area to wall area as well as window location be examined. Obviously, in considering a building design, it is important to take into consideration the external environment. Most occupants will not want to live in a totally enclosed cell. This requires that windows be used properly for enhancing views and light levels, and for entrance regions. However, long unbroken walls of glass waste energy and should be avoided. Horizontal windows elevated to normal heights should be considered. They provide adequate light, undiminished views, and minimize glass area to reduce heat losses and gains.

EXAMPLE 9-1

Table 9-4 lists the materials in the construction of a typical wall. With an uninsulated wall, the total resistance is 4.4 and the U factor is 0.22. With R-11 insulation, total wall resistance is 14.43 and the U factor is 0.07. The R-11 insulation reduces the heat loss and gain through the wall by a factor of three.

TABLE 9-4. U Factors for a Wall

Wall Construction	Uninsulated Wall Resistance	Insulated Wall Resistance
Outside Surface	0.17	0.17
Wood Bevel Siding, Lapped	0.81	0.81
1/2" Insulation Board Sheathing, Regular Density	1.32	1.32
3 1/2" Air Space	1.01	----
R-11 Insulation	----	11.00
1/2" Gypsum Board	0.45	0.45
Inside Surface	0.68	0.68
TOTAL	4.44	14.43

$U = \frac{1}{R} = \frac{1}{4.44} = 0.22$

$U = \frac{1}{R} = \frac{1}{14.43} = 0.07$

TABLE 9-5. Window Heat Losses

Infiltration Around a Window:

3' x 5' Double-hung, non-weather-stripped, wood window, average installation

From Table 2, Chapter 19, ASHRAE Guide

AIR LEAKAGE RATE = 27 ft^3/ft-crack-hour (1/16" crack, 3/64" clearance)

$L = (2 \times 5') + (3 \times 3')$ [illegible] 19 ft.

$Q = 0.018\ BL\ (T_i - T_o)$

$Q = (0.018)\ (27)\ (19)\ (70-0) = 646$ Btu/hr.

TRANSMISSION LOSSES:

Single Glaze $U = 1.13$ Btu/(hr) (ft^2) (^oF)

$Q = UA\ (T_i - T_o) = (1.13)\ (15)\ (70-0) = 1{,}187$ Btu/hr.

WINDOW TOTAL LOSS = 1,833 Btu/hr.

Consider a wall area of 374 square feet and a temperature differential between the inside and outside of 70°F. The heat loss through the insulated wall is 1833 Btu/hr and the heat loss for an uninsulated wall is 5760 Btu/hr.

EXAMPLE 9-2

Consider the losses through a window. Table 9-5 presents information for a typical window. The U factor for a single glazed window from Table 5-4 is 1.13. A 15-square-foot window with a 70° temperature differential results in 1187 Btu/hr heat loss. Table 9-5 also includes computation for infiltration around the window using the "crack" method. The infiltration losses for an average window installation is 646 Btu/hr, for a total window loss of 1833 Btu/hr. In other words, 15 square feet of window area has the same loss as 374 square feet of insulated wall or 120 square feet of uninsulated wall. One square foot of glass has the same heat loss as 25 square feet of insulated wall area in example 9-1.

Loss through windows can be reduced by using double glass windows. From values given in Table 5-4, the transmission loss could be reduced from 1.13 to 0.65 resulting in 40-percent reduction in total loss.

Other building design factors include the shading of windows in the summertime to prevent radiation gain. In wintertime operation the heat gain from the sun can be advantageous. Both summer shading and winter illumination can be accomplished by using properly dimensioned overhangs or external shades. External shading is preferable to internal shading because it keeps energy out of the building. While internal shading is effective, not all of the heat can be conducted and radiated out once it has entered. Buffered entryways, or air-locks, reduce infiltration losses when a small buffered area is provided between doorways. As noted from the example in Module 5, infiltration is a major factor in heat losses and gains for a building.

Proper sizing of the heating and cooling system for the building is important. The combustion efficiency of a gas-fired furnace can be reduced by 15 to 20 percent below its steady-state operating efficiency if there is frequent cycling. The more oversized the furnace is, the more intermittent will be its operation and the less efficient will be its operation. A sampling of on-time versus degree-day heating obtained for typical gas-fired furnaces in the Fort Collins area indicates that most systems are oversized by a factor of at least 2. This has been accepted in the past because sizing of heating units has been by "rule of thumb" rather than by calculation of heat losses as outlined in Module 5. The occupants have been satisfied because the cost for operating has been very low.

BUILDING OPERATION

The effect of building operation on energy conservation is more difficult to predict because it is influenced by the life styles and the living habits of the occupants. Energy usage of nearly identical buildings can vary by as much as a factor of 2 because some people prefer to maintain room temperatures of 75 to 78°F in the wintertime, while others prefer lower internal temperatures of 64 to 68°F.

Studies have shown that utilizing clock thermostats or manual thermostatic set-back, that is, turning "down" the thermostat between the hours of 10 P.M. and 7 A.M., will result in energy savings. Each degree of thermostatic set-back will save between 1½ to 2½ percent in energy use for heating. For a 7-degree thermostatic set-back, from 75 to 68°F, or 68 to 61°F during night-time ~~hours~~, savings of 10 to 17 percent can be achieved.

In order to effect energy savings through lower thermostat settings and set-backs, an effective educational campaign will have to be carried out, to convince the occupant that significant amounts of his money can be saved. It should be pointed out to home-owners or occupants of buildings that prudent operating practices are critical to reduce overall energy usage.

BUILDING MAINTENANCE

The obvious maintenance item concerns the heating and cooling equipment which needs to be kept to near-peak efficiency. In addition to the heating/cooling equipment maintenance, the building structure should be maintained. For example, through settling, walls may move away from fireplaces, resulting in larger infiltration losses. Fireplaces are heat wasters, but where they are desired, combustion air should be drawn in from the outside. The building exterior should be painted at regular intervals to maintain the thermal characteristics of the surface as well as to preserve the material.

COST-PERFORMANCE TRADE-OFFS

In the examples considered in this section of the module, the main thesis of the proposed design analysis for houses is that the annual heat flow through the exterior surface of the building is directly proportional to the annual number of degree days of heating. Although the method is not exact, it is based on acceptable correlations over long periods of operation using typical houses, and it should be well within reasonable uncertainty limits for the home designer. Moreover, the method should be applicable to other buildings which do not have a high annual internal heat generation rate relative to the annual heat load.

Consider the house plan shown in Figure 9-1. The house is a simple two-story house and is to be located in a region where the winter heat load is 6000 degree days. In the example, assume that the house is to be built with no wall insulation and with only R-7 insulation in the ceiling. The question to be addressed is "should the ceiling insulation be increased to R-30?" Is this design change cost-effective?

Table 9-6 presents heat loss sensitivity results for the house considered above. Changes to the basic house are listed as well as the resulting heat loss. The design ambient temperature was -10°F and the indoor temperature was chosen as 68°F. For the original design, the calculated heat load is 88,846 Btu/hr for the design temperature condition, as shown on the first line of Table 9-6.

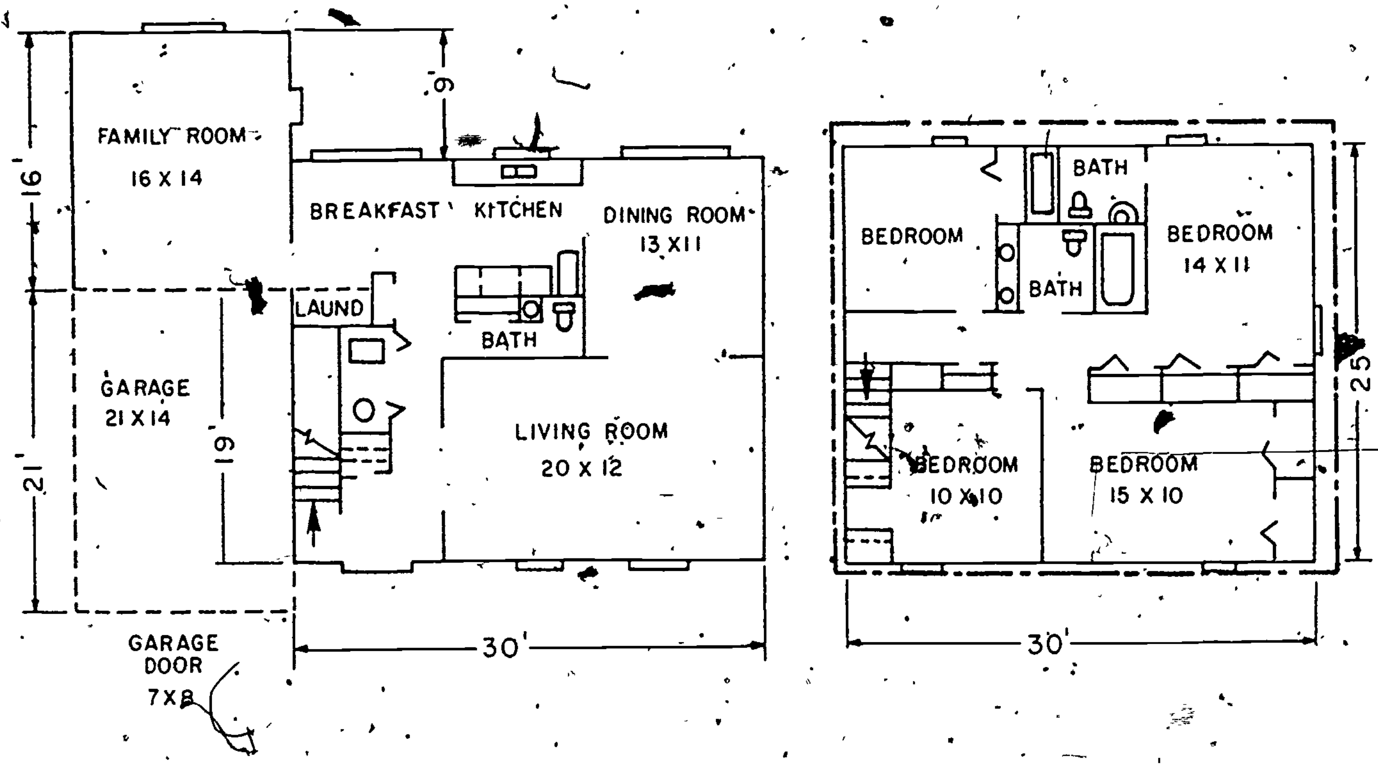
If the outdoor temperature remains -10°F all day (24 hours) the degree days is

$$DD = 65°F - (-10°F) = 75°F \text{ days.}$$

The heating load for the building is therefore

$$\frac{88,846}{75} \times 24 = 28,430 \text{ Btu/DD}$$

If the outdoor temperature varies, from a maximum of 30°F to a minimum of -10°F, the degree days is determined


FAMILY ROOM
16 X 14
16'
9
BREAKFAST
KITCHEN
DINING ROOM
13 X 11
LAUND
BATH
GARAGE
21 X 14
19
21'
LIVING ROOM
20 X 12
GARAGE
DOOR
7X8
30'
BATH
BEDROOM
BEDROOM
14 X 11
BATH
BEDROOM
10 X 10
BEDROOM
15 X 10
25
30'

Areas		Sq. Ft.
Ceiling Area		1621.0
Wall		2336.0
Windows:	North	45.0
	West	50.5
	South	100.0
	East	0.0
		195.5
Door		98.0
Roof		1793.0

Figure 9-1. Example House

TABLE 9-6 HEAT LOAD SIMULATION RESULTS

WALL INSULATION	CEILING INSULATION	DOORS	WINDOWS	BTUH HEAT LOSS
R-0	R-7	Solidwood 1.5 in.	Single glass 100% glass	88,846
R-0	R-7	Solidwood 1.5 in.	Single glass 80% glass	87,234
R-0	R-7	Solidwood 2.0 in.	Single glass 80% glass	87,073
R-0	R-7	Storm metal & 1.5 in. Solidwood	Single glass 80% glass	86,989
R-0	R-7	Solidwood 1.5 in.	Double insulating Double glass 80% 3/16 in. air space	82,077
R-0	R-7	Solidwood 2.0 in.	Double insulating Double glass 80% 3/16 in. air space	81,916
R-0	R-7	Storm metal & 1.5 in. Solidwood	Double glass 80% Double insulating 3/16 in. air space	81,832
R-0	R-7	Solidwood 1.5 in.	Triple insulating Triple glass 80% 1/2 in. air space.	77,605
R-0	R-7	Storm metal & 2.0 in. Solidwood	Triple insulation Triple glass 80% 1/2 in. air space	77,229
R-7	R-7	Solidwood 1.5 in.	Single glass 100% glass	66,636
R-7	R-7	Solidwood 1.5 in.	Double insulating Double glass 80% 3/16 in. air space	59,867
R-7	R-7	Storm metal & 1.5 in. Solidwood	Triple insulating Triple glass 80% 1/2 in. air space	55,149
R-7	R-11	Solidwood 1.5 in.	Single glass 80% glass	62,500
R-7	R-11	Storm metal & 1.5 in. Solidwood	Single glass 80% glass	62,255

TABLE 9-6 HEAT LOAD SIMULATION RESULTS (continued)

WALL INSULATION	CEILING INSULATION	DOORS	WINDOWS	BTUH HEAT LOSS
R-7	R-11	Solidwood 1.5 in.	Double insulation Double glass 80% 3/16 in. air space	57,343
R-7	R-11	Storm metal & 1.5 in. Solidwood	Double insulating Double glass 80% 3/16 in. air space	57,102
R-7	R-11	Storm metal & 2.0 in. Solidwood	Triple insulating Triple glass 80% 1/2 in. air space	52,496
R-11	R-11	Solidwood 1.5 in.	Single glass Double glass 80% 3/16 in. air space	58,987
R-11	R-11	Storm metal & 1.5 in. Solidwood	Double insulating Double glass 80% 3/16 in. air space	53,585
R-11	R-11	Storm metal & 2.0 in. Solidwood	Double insulating Single glass 80% Emissivity = 0.2 1/2 in. air space	49,525
R-11	R-11	Storm metal & 2.0 in. Solidwood	Triple insulating Triple glass 80% 1/2 in. air space	48,983
R-11	R-19	Solidwood 1.5 in.	Single glass 100% glass	58,233
R-11	R-19	Solidwood 1.5 in.	Single glass 80% glass	56,621
R-11	R-19	Solidwood 1.5 in.	Double insulating Double glass 80% 3/16 in. air space	51,464
R-11	R-19	Solidwood 1.5 in.	Double insulating Single glass 80% Emissivity = 0.2 1/2 in. air space	47,534
R-11	R-19	Storm metal & 1.5 in. Solidwood	Double insulating Single glass 80% Emissivity 80% 1/2 in. air space	47,288
R-11	R-19	Storm metal & 2.0 in. Solidwood	Triple insulating Triple glass 80% 3/16 in. air space	46,616

TABLE 9-6 HEAT LOAD SIMULATION RESULTS (continued)

WALL INSULATION	CEILING INSULATION	DOORS	WINDOWS	BTUH HEAT LOSS
R-0	R-11	Wooden Door 1.5 in.	Single glass 100% glass	86,303
R-0	R-19	Wooden Door 1.5 in.	Single glass 100% glass	83,915
R-7	R-0	Wooden Door 1.5 in.	Single glass 100% glass	84,698
R-7	R-11	Wooden Door 1.5 in.	Single glass 100% glass	64,112
R-7	R-19	Wooden Door 1.5 in.	Single glass 100% glass	61,743
R-11	R-0	Wooden Door 1.5 in.	Single glass 100% glass	81,165
R-11	R-7	Wooden Door 1.5 in.	Single glass 100% glass	63,120
R-11	R-11	Wooden Door 1.5 in.	Single glass 100% glass	60,599
R-19	R-0	Wooden Door 1.5 in.	Single glass 100% glass	77,748
R-19	R-7	Wooden Door 1.5 in.	Single glass 100% glass	59,720
R-19	R-11	Wooden Door 1.5 in.	Single glass 100% glass	57,202
R-19	R-19	Wooden Door 1.5 in.	Single glass 100% glass	54,838
R-0	R-0	Wooden Door 1.5 in.	Single glass 100% glass	107,032
R-19	R-40	Wood Storm Doors & 2.0 in. Solidwood Doors	Triple insulating 1/2 in. air space & Storm windows 60% glass-wood sash	40,079

Total Window Area = 195.42 Sq. Ft.
Total Door Area = 98 Sq. Ft.
45° Pitched Roof
No Basement

TABLE 9-6 HEAT LOAD SIMULATION RESULTS (continued)

WALL TYPE + INSULATION	ROOF TYPE + INSULATION	DOORS	WINDOWS	BTUH HEAT LOSS
Frame +R-7	45-degree Pitched Roof +R-0	Wooden Door 1.5 in.	Single glass 100% glass	84,698
Frame +R-7	Flat Roof Wood Construction +R-0	Wooden Door 1.5 in.	Single glass 100% glass	74,981
Frame +R-7	Flat Roof Metal Construction +R-0	Wooden Door 1.5 in.	Single glass 100% glass	73,361
Frame +R-0	45-degree Pitched Roof +R-7	Wooden Door 1.5 in.	Single glass 100% glass	88,846
Frame +R-7	45-degree Pitched Roof +R-7	Wooden Door 1.5 in.	Single glass 100% glass	66,636
Brick and Frame Partition +R-0	45-degree Pitched Roof	Wooden Door 1.5 in.	Single glass 100% glass	97,124
Brick and Frame Partition +R-7	45-degree Pitched Roof	Wooden Door 1.5 in.	Single glass 100% glass	94,324
Brick + Cinder and Frame Partition +R-0	45-degree Pitched Roof	Wooden Door 1.5 in.	Single glass 100% glass	79,183
Brick + Cinder and Frame Partition +R-7	45-degree Pitched Roof	Wooden Door 1.5 in.	Single glass 100% glass	76,501

Total Window Area = 195.42 Sq. Ft.
Total Door Area = 98 Sq. Ft.
No Basement

$$DD = 65 - [\frac{30 + (-10)}{2}] \quad \text{or} \quad DD = 55°F \text{ days.}$$

The heat load for the day would then be

$$28{,}430 \times 55 = 1{,}564{,}000 \text{ Btu.}$$

The load for the entire heating season, given 6000 DD, is

$$L = 28{,}430 \frac{\text{Btu}}{\text{DD}} \times 6000 \frac{\text{DD}}{\text{year}} = 170{,}600{,}000 \frac{\text{Btu}}{\text{year}}$$

Now, let us consider the cost of energy. Assume, for example, that the house is to be electrically heated, with electricity cost at $0.03 per kwh. The cost of electricity per million Btu is as given below:

$$\frac{1{,}000{,}000 \text{ Btu} \times \$0.03}{3413 \text{ (Btu)(kwh)}^{-1}} = \$8.79 \text{ per million Btu.}$$

The efficiency of an electric resistance heater is 1.0, and so the cost of heating per delivered million Btu is $8.79. The load times the cost per million Btu results in a heating cost, C, of $1500 per year.

$$C = \$8.79 \times 170.6 = \$1500/\text{year.}$$

If the building is modified in design for ceiling insulation of R-19 rather than R-7, everything else remaining the same, the load is 83,915 Btu as shown on the third page of Table 9-6. Making similar calculations, as the foregoing, the heating cost per year would be $1416. This is a net savings of $84 per year.

$$\frac{83{,}915 \text{ Btu}}{75} \times 24 = 26{,}850 \text{ Btu/DD}$$

$$L = 26{,}850\ \frac{\text{Btu}}{\text{DD}} \times 6000\ \frac{\text{DD}}{\text{year}} = 161{,}100{,}000\ \text{Btu/year}$$

$$C = \$8.79/\text{million Btu} \times 161.1\ \text{million}\ \frac{\text{Btu}}{\text{year}} = \$1416/\text{year}$$

$$\text{Savings} = \$1500 - \$1416 = \$84/\text{year.}$$

The savings factor is defined as

$$SF = \frac{\text{savings}}{\text{annual cost of energy}}$$

For this example

$$SF = \frac{84}{1500} = 0.056$$

Is the savings cost worth the increased cost in installing R-19 insulation over R-7? To answer this question we must resort to some educated guesses.

Assume that the mortgage lifetime is 25 years and the interest rate is 9%, and that energy costs over those twenty-five years increase at the rate of 6% a year. A 6%-per-year increase indicates that the energy cost in the twenty-fifth year will be about 4.3 times the present value, or $37.80 per million Btu. Although this seems large, current estimates tend to show increases in fuel costs of about 3 to 5 times the current value.

Figure 9-2 shows curves of maximum first-cost investments that can be made for various fuel cost savings and energy rate increases at different savings factors and interest rates for a mortgage period of 25 years. The curves are based on the equation,

$$MFC = SF\left(\frac{a(a^n-1)}{a-1}\right) \times PEE\ , \tag{9-1}$$

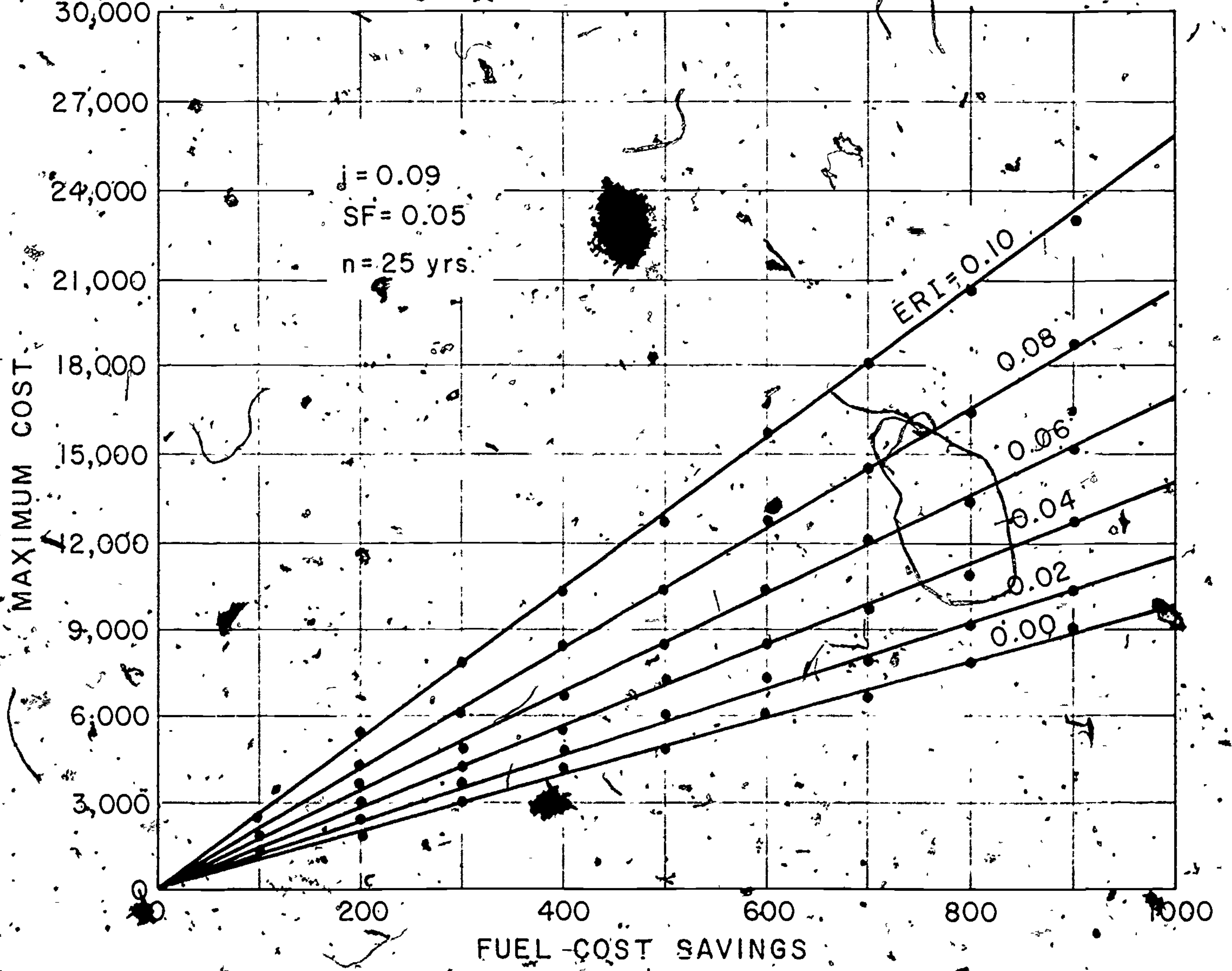


Figure 9-2. Fuel Cost Savings

in which $a = \frac{1+ERI}{1+i}$

and MFC is maximum first cost,

SF is savings in fuel,

ERI is energy rate increase, decimal,

i is interest rate, decimal,

n is number of years,

PEE is present energy expense.

From Figure 9-2, with the energy savings of $84, and ERI of 0.06, the maximum that should be expended is ~$1400. In other words, for this case, if it costs less than $1400 for the materials and the installation, then using R-19 over R-7 insulation is cost beneficial and should be incorporated. A change in conditions can alter results; for example, a change in the estimated inflationary rate of energy from 6 percent to 2 percent results in a maximum cost reduction from $1400 to $908. A 2-percent inflationary increase over the next twenty-five years yields a 60-percent increase in the cost of energy over current costs, a number certainly lower than most experts have projected. The 6% rate (a four- or five-fold increase in the cost of energy over the next twenty-five years) seems to be in reasonable agreement with the projections of many experts.

Similarly, we can analyze the effects of other changes once we have the capability to calculate heat load information. Consider a 20-percent reduction of window area from that shown in the plans. From Table 9-6, the heat loss rate is about 1600 Btu/hr less. Another change which could be considered is adding wall insulation. The inclusion of R-9 wall insulation with increased ceiling insulation could be evaluated. All of these energy-saving comparisons can be made by following the procedure given in the foregoing example.

These calculations are predicated on the ability to make heat load calculations following procedures as outlined in Module 5. A computerized technique is convenient to determine ~~the~~ sensitivity of the heat loss to design changes. Once the effect has been assessed in terms of heat losses, it is a relatively simple matter to make cost analyses.

It is interesting to note from Table 9-6 that with R-19 wall insulation, R-47 ceiling insulation, storm doors, and double glass windows with storm windows (an extremely well-designed building thermally), the heat load can be reduced by over 50 percent. This would indicate that savings of 55 or 60 percent of the original energy bill are possible by better building designs. For design of a solar heated and cooled building it is necessary to achieve the best over-all building design consistent with cost to minimize the size of the solar system. A combination of energy-conserving measures should be considered early in the design of a new building.

REFERENCE

"Energy Conservation Computer Software" distributed by General Services Administration, Washington, D.C.

TRAINING COURSE IN
THE PRACTICAL ASPECTS OF
DESIGN OF SOLAR HEATING AND COOLING SYSTEMS
FOR
RESIDENTIAL BUILDINGS

MODULE 10

DETAILED DESIGN CALCULATIONS

SOLAR ENERGY APPLICATIONS LABORATORY
COLORADO STATE UNIVERSITY
FORT COLLINS, COLORADO

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LIST OF SYMBOLS

DD	Degree Day
f	Fraction of monthly heating load supplied by the solar system
F	Fraction of annual heating load supplied by the solar system
F_R	Collector plate efficiency factor
$\overline{H}$	Daily average solar radiation for the month on horizontal surface
$\overline{H}_o$	Daily average solar radiation on horizontal surface outside the earth's atmosphere
$\overline{H}_T$	Daily average solar radiation, over a month, on a tilted surface
$\overline{K}_T$	Ratio, $\overline{H}/\overline{H}_o$
L_H	Heating load, Btu
L	Total load, Btu
$\overline{R}$	Ratio, $\overline{H}_T/\overline{H}$
S	Total solar radiation for month
SHW	Service Hot Water
$\overline{T}_A$	Average daily temperature $(T_{high} + T_{low})/2$
T_{REF}	Reference temperature 212 ^oF
U_L	Solar collector overall heat loss coefficient, Btu/(hr)(ft^2)(^{o}F)
$\overline{\tau\alpha}$	Transmittance absorptance product average for collector; transmittance through covers, absorptance by plate

INTRODUCTION

TRAINEE-ORIENTED OBJECTIVE

To practice the calculations for predicting performance of a solar system.

SUB-OBJECTIVES

1. To learn to use the f-charts for design purposes.
2. To examine the relationship between economics and energy-conservation measures.

PROBLEMS

1. Consider the same house that was used in the previous design laboratory. Determine the size for a solar system (liquid type) to provide space heating and service hot water loads. The collector to be used has the following properties:

$$F'_R \overline{\tau\alpha} = 0.724$$

$$F'_R U_L = 0.947 \text{ Btu/Hr-Ft}^2 \, {}^oF$$

Use the worksheets provided in the notes. The collector tilt is to be latitude + 15^0. Compare the results obtained in this design exercise with those obtained in the previous design session.

2. Now suppose that the heat load of the house is decreased from 17,200 Btu/DD to 16,000 Btu/DD by adding insulation at a cost of $300. Repeat problem 1.

CLIMATOLOGICAL DATA (DENVER)

(1) MONTH	(2) $\overline{H}$ Btu/FT²-DAY	(3) $\overline{K}_T$	(4) $\overline{R}$	(5) $\overline{H}_T$ Btu/FT²-DAY	(6) DD	(7) $\overline{T}_A$ °F	(8) $T_{REF}-\overline{T}_A$	(9) L_H Btux10^{-6}	(10) L Btux10^{-6}	(11) S Btu/FT²-MO.
JAN	742	.56			1132	28	174			
FEB	989	.55			938	32	170			
MAR	1480	.62			887	36	166			
APRIL	1697	.56			558	46	156			
MAY	1697	.49			288	56	146			
JUNE	1937	.53			66	63	139			
JULY	1919	.54			6	65	137			
AUG	1620	.51			9	65	137			
SEPT	1520	.58			117	61	141			
OCT	1144	.58			428	51	151			
NOV	819	.57			819	38	164			
DEC	672	.56			1035	32	170			
	F3-1, 3-12	$\overline{H}/\overline{H}_o$ - $\overline{H}_o$ T3-2	F3-35	$\overline{H}_T = \overline{H}\overline{R}$	T4-1	$T_\Delta = 65-\frac{DD}{n}$	202 - (7)	$L_H = Q_{DES} \times (6)$	L = (9) = SHW	(5) x n

n = number of days in given month

SHW = load (gallons/day) x (8.33 lb/gal) x (1 Btu/lb - °F) x (ΔT °F) x (n days)

CLIMATOLOGICAL DATA

MONTH	$\overline{H}$ Btu/FT^2-DAY	$\overline{K}_T$	$\overline{R}$	$\overline{H}_T$ Btu/FT^2-DAY	DD	$\overline{T}_A$ °F	$T_{REF}-\overline{T}_A$	L_H Btux10^{-6}	L Btux10^{-6}	S Btu/FT^2-MO.
JAN										
FEB										
MAR										
APRIL										
MAY										
JUNE										
JULY										
AUG										
SEPT										
OCT										
NOV										
DEC										

CLIMATOLOGICAL DATA

MONTH	$\overline{H}$ Btu/FT^2-DAY	$\overline{K}_T$	$\overline{R}$	$\overline{H}_T$ Btu/FT^2-DAY	DD	$\overline{T}_A$ °F	$T_{REF}-\overline{T}_A$	L_H Btux10^{-6}	L Btux10^{-6}	S Btu/FT^2-MO.
JAN										
FEB										
MAR										
APRIL										
MAY										
JUNE										
JULY										
AUG										
SEPT										
OCT										
NOV										
DEC.										

CLIMATOLOGICAL DATA

MONTH	$\overline{H}$ Btu/FT^2-DAY	$\overline{K}_T$	$\overline{R}$	$\overline{H}_T$ Btu/FT^2-DAY	DD	$\overline{T}_A$ °F	$T_{REF}-\overline{T}_A$	L_H Btux10^{-6}	L Btux10^{-6}	S Btu/FT^2-MO.
JAN										
FEB										
MAR										
APRIL										
MAY										
JUNE										
JULY										
AUG										
SEPT										
OCT										
NOV										
DEC										

FRACTION OF LOAD SUPPLIED BY SOLAR

$F_R U_L$ Δt/L* $(T_{REF}-\bar{T}_A)$ $FT^{-2}x10^{-3}$	$F_R{}^1\overline{\tau\alpha}$ S/L $FT^{-2}x10^{-4}$	X & Y VALUES						f - FRACTION OF LOAD SUPPLIED BY SOLAR		
		AREA =		AREA =		AREA =		AREA	AREA	AREA
		X	Y	X	Y	X	Y			

FRACTION OF LOAD SUPPLIED BY SOLAR

$F_R U_L \Delta t/L^*$ $(\bar{T}_{REF}-\bar{T}_A)$ $FT^{-2}x10^{-3}$	$F_R{}^1 \overline{\tau\alpha}$ S/L $FT^{-2}x10^{-4}$	X & Y VALUES						f - FRACTION OF LOAD SUPPLIED BY SOLAR		
		AREA =		AREA =		AREA =		AREA	AREA	AREA
		X	Y	X	Y	X	Y			

FRACTION OF LOAD SUPPLIED BY SOLAR

$F_R U_L \Delta t/L^* (T_{REF}-\bar{T}_A)$ $FT^{-2} x 10^{-3}$	$F_R' \overline{\tau\alpha} S/L$ $FT^{-2} x 10^{-4}$	X & Y VALUES						f - FRACTION OF LOAD SUPPLIED BY SOLAR		
		AREA =		AREA =		AREA =		AREA	AREA	AREA
		X	Y	X	Y	X	Y			

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TRAINING COURSE IN
THE PRACTICAL ASPECTS OF
DESIGN OF SOLAR HEATING AND COOLING SYSTEMS
FOR
RESIDENTIAL BUILDINGS

MODULE 11

COLLECTORS

SOLAR ENERGY APPLICATIONS LABORATORY
COLORADO STATE UNIVERSITY
FORT COLLINS, COLORADO

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INTRODUCTION

TRAINEE-ORIENTED OBJECTIVE

The objective in this module is to describe the elements of fluid-heating solar collectors and identify important parameters which affect system designs and performance.

SUB-OBJECTIVES

From this module the trainee should be able to:

1. Define collector efficiency
2. Identify the parameters which affect collector efficiency
3. Recognize the advantages and disadvantages of air-heating and liquid-heating collectors
4. Identify and select the solar collectors best suited to meet specific requirements.

TYPES OF COLLECTORS

Collectors are divided into two classes, liquid-heating and air-heating solar collectors. Both types consist of an absorber plate with black surface coating contained in a metal frame box with one or more transparent covers above the absorber plate. The covers are transparent to incoming solar radiation and relatively opaque to outgoing (long-wave) radiation, but their principal purpose is to reduce heat losses by convection. Insulation is used to reduce conduction heat losses through the back of the collector, or a vacuum jacket may be employed to reduce both conduction and convection heat losses from the absorber surface. Although nearly all practical systems for solar space heating and hot water heating involve the use of flat-plate absorbers, tubular absorbers inside evacuated glass tubes or at the focus of some type of concentrating device (lens or mirror) have been built and tested.

Typical Liquid Collector

Figure 11-1 is a partially sectioned diagram of a typical flat-plate liquid solar collector. The drawing shows a commercially manufactured collector comprising a glass-covered metal box containing an absorber plate to which an array of tubes is attached, beneath which insulation is provided. A liquid is pumped through the collector tubes and manifolds for heating. Typical collector dimensions are 6.5 ft by 3 ft by 6 inches. The space between glass covers is about one-half inch and the inner glass cover is about one inch above the absorber plate. Two to four inches of insulation such as heat-resistant fibrous glass are commonly used below the absorber plate. Metal is probably the best material for absorber plates, and good thermal contact is required between the absorber plate and the tube through which the liquid is transported. Volumetric flow rate is typically 0.02 gal/min. ft^2 of collector surface area.

Typical Air Collector

Figure 11-2 is a diagrammatic sketch of a typical air-heating solar collector. The principal difference between the air and liquid types of collectors is the size and configuration of the fluid conduits. The figure shows three wide air passages directly beneath the absorber plate. Air, therefore, flows in contact with nearly the entire absorber surface, for effective heat transfer. The design shown also has internal manifolds for air distribution to all collector panels in a close-fitting array. Volumetric flow rate is typically 2 cfm/ft^2 of collector surface area.

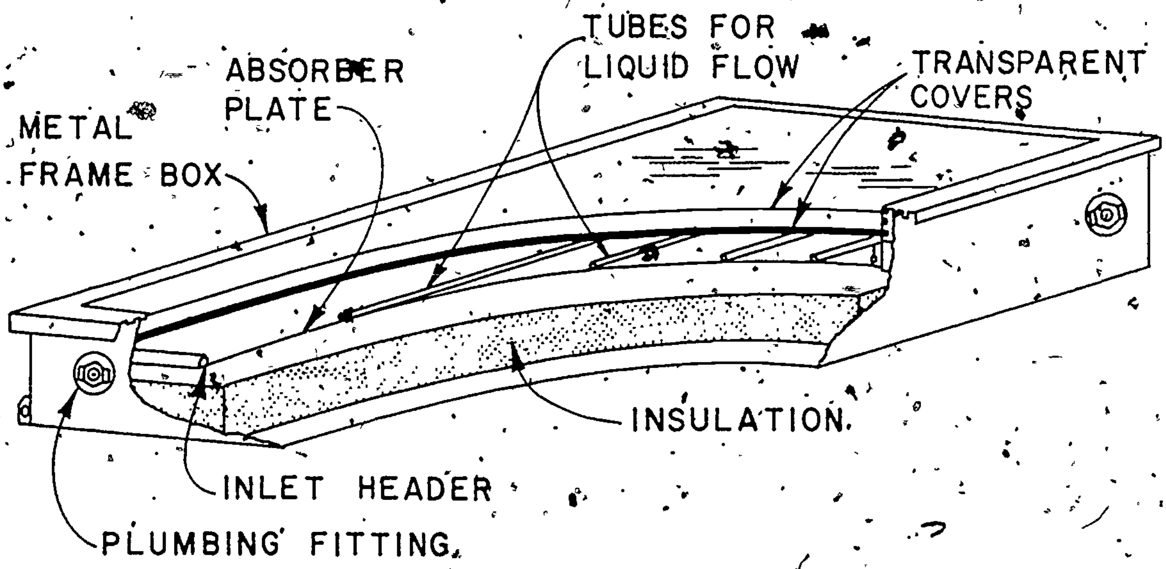

TUBES FOR LIQUID FLOW
ABSORBER PLATE
TRANSPARENT COVERS
METAL FRAME BOX
INSULATION
INLET HEADER
PLUMBING FITTING

Figure 11-1. Typical Liquid-Heating Collector

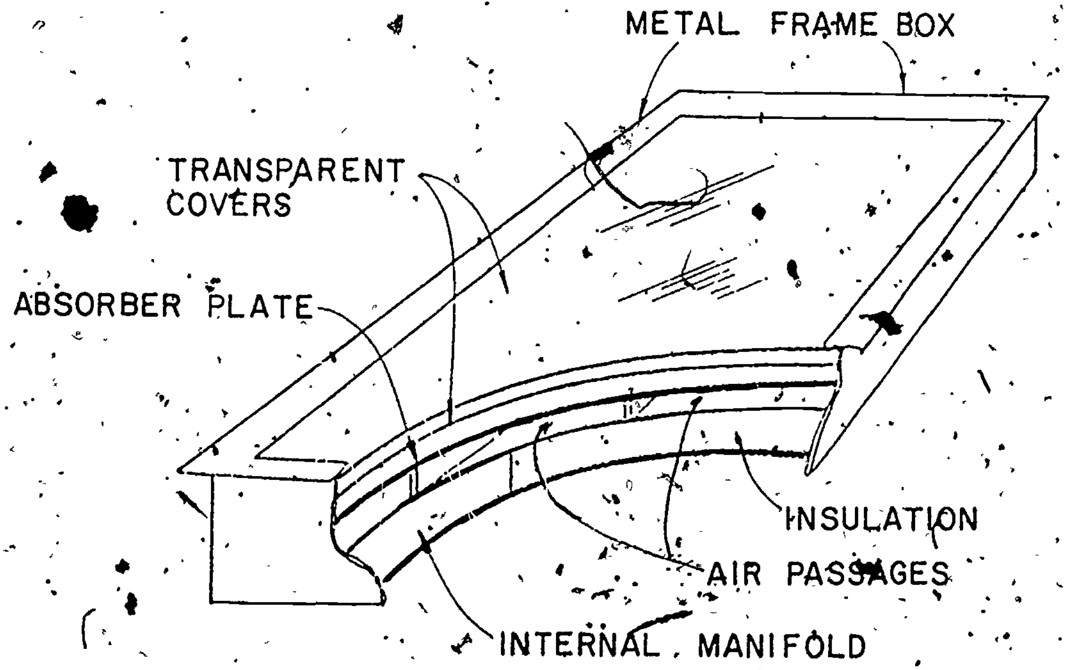

METAL FRAME BOX
TRANSPARENT COVERS
ABSORBER PLATE
INSULATION
AIR PASSAGES
INTERNAL MANIFOLD

Figure 11-2. Typical Air-Heating Collector

GENERAL PRINCIPLES

A solar collector is a device for converting the energy in solar radiation to heat in a fluid. This conversion is accomplished by absorbing the solar radiation on a broad, thin metal surface which is in contact with a stream of liquid or gas. Absorption of solar energy causes the temperature of the metal surface to rise so that the temperature of the fluid increases as it moves past the surface.

Under steady conditions, the useful heat delivered by the solar collector is equal to the energy absorbed in the metal surface minus the heat losses from that surface directly and indirectly to the surroundings. This principle can be stated in the relationship:

$$Q_u = A_c \left[H_T \tau\alpha - U_L (\bar{T}_p - T_a)\right] \qquad (11\text{-}1)$$

where

Q_u is useful energy delivered by collector, Btu/hour

A_c is total collector area, ft^2

H_T is the solar energy received on the upper surface of the sloping collector structure, $Btu/hr \cdot ft^2$ of tilted surface.

τ is fraction of the incoming solar radiation which reaches the absorbing surface, no dimensions

α is fraction of the solar energy reaching the surface which is absorbed, absorptivity, no dimensions

U_L is the overall heat loss coefficient, Btu, transferred to the surroundings per hour/ft^2 of exposed collector surface per degree difference between average collector surface temperature and the surrounding air temperature

T_p is average temperature of the upper surface of the absorber plate, ^oF

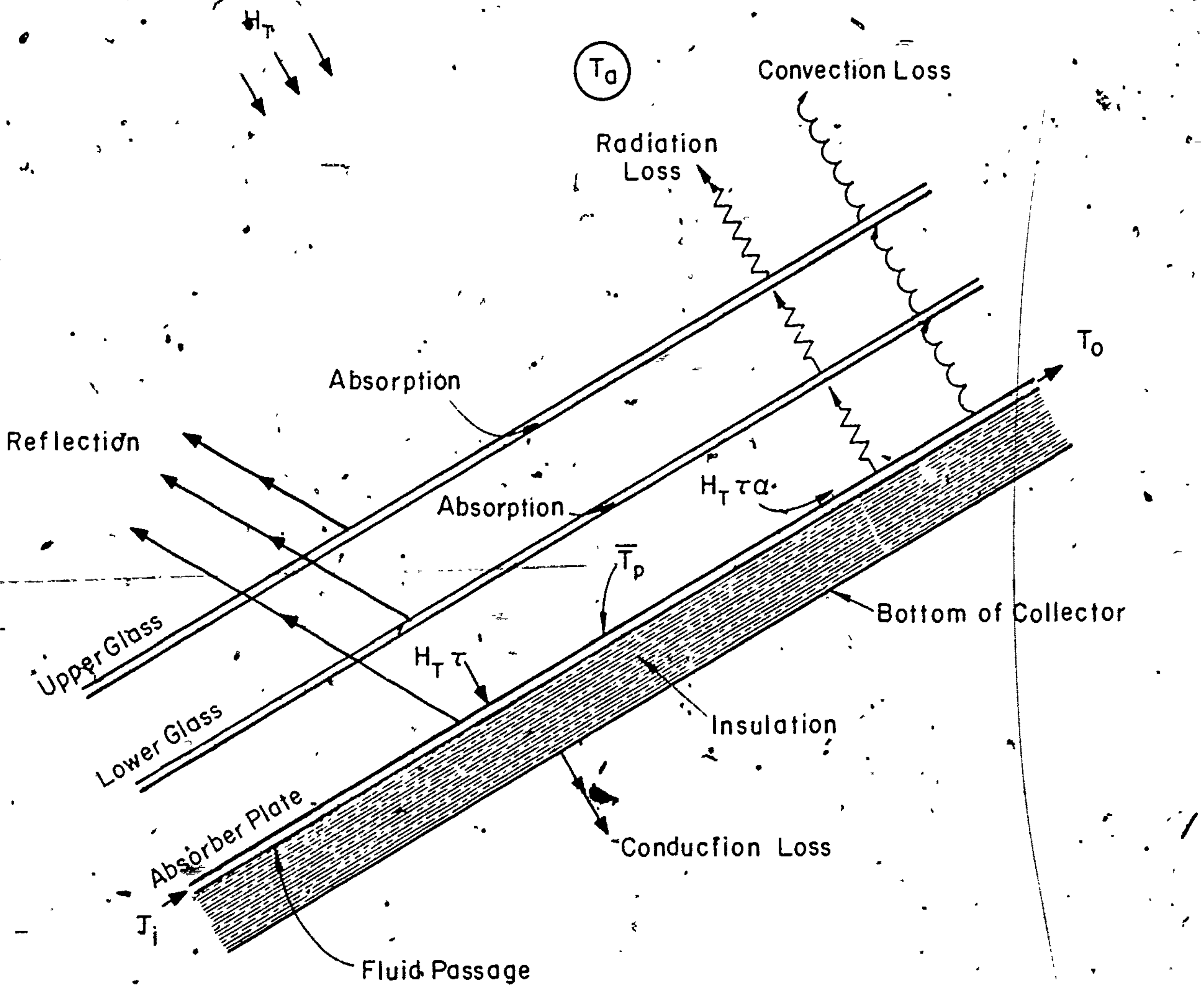
T_a is atmospheric temperature, ^oF.

A diagrammatic representation of the terms in this relationship is shown in Figure 11-3.

HEAT LOSSES FROM COLLECTOR

In order that the performance of the collector can be as high as economically practical, the design and operating factors which can maximize the value of the first term on the right hand side of Equation (11-1) and can minimize the value of the second term are selected. In other words, the greater the energy absorption in the metal surface and the lower the heat loss from that surface, the higher will be the useful recovery. If a bare metal plate serves as the collector, and with typical values of 2 to 10 Btu/hr·ft^2·^oF for the coefficient of heat transfer to the atmosphere (U_L), the rates of heat loss will be large, so that an absorber plate temperature of 25 to 50 degrees above atmospheric temperature would be the maximum achievable under typical full solar radiation of 300 Btu/hr·ft^2. Under these conditions no useful heat would be delivered from the collector because the heat loss would be equal to the solar heat absorbed, leaving nothing for useful delivery.

To reduce the rate of heat loss occurring by radiation and convection, one or more transparent surfaces, such as glass, can be placed above the metal surface. The glass will transmit as much as 90 percent of the solar radiation striking it, and it will greatly reduce the heat loss coefficient, U_L. This reduction is due to the suppression of convection losses by the relatively stagnant air layer between the absorber plate and the glass, and by intercepting the long-wave, thermal radiation emitted by the hot metal


H_T
T_a
Convection Loss
Radiation Loss
T_o
Absorption
Reflection
Absorption
$H_T \tau \alpha$
$\overline{T}_p$
Bottom of Collector
Upper Glass
$H_T \tau$
Insulation
Lower Glass
Absorber Plate
Conduction Loss
T_i
Fluid Passage

Absorbed Energy $= A_c \cdot H_T \, \tau\alpha$

Effective Heat Loss $= A_c \, U_L(\overline{T}_p - T_a)$

Figure 11-3. Definition Sketch for Equation (11-1).

surface because glass is opaque to the long-wave radiation. The heat loss coefficient can be reduced to 1 to 2 Btu/hr·ft^2·°F by the use of one glass cover. Similar benefits can be realized by use of certain transparent plastic materials.

Further reduction in the heat-loss coefficient can be realized by using a second transparent surface with an air space between the two surfaces. Two relatively stagnant air barriers to convection loss are then present, as well as two surfaces impeding radiation loss. Coefficients in the range of 0.7 Btu/hr·ft^2·°F are typically then obtained.

Radiation losses can be reduced by other techniques, such as by reducing the radiation-emitting characteristics of the heat-absorbing surface. This measure is discussed in the section pertaining to the solar radiation absorbing characteristics of the surface. Thermal radiation emitted by the absorber plate may also be reduced by reflecting it downward from the lower glass cover by employing an infrared-reflecting coating on the glass. An optically transparent, very thin layer of tin oxide or indium oxide deposited on the glass will reduce radiation loss by reflecting it back to the absorber plate. This coating absorbs a small fraction of the solar radiation, however, so the reduced thermal loss is largely offset by reduced solar energy input to the absorber plate.

Significant losses can occur from the side and back of the collector unless insulation is used. It is advisable to use a high-temperature insulation adjacent to the back side of the absorber plate layered with a lower temperature insulation to provide the required resistance to heat flow. The total R value of the insulation should be at least 10 for medium-temperature flat-plate collectors.

A transparent honeycomb of thin plastic film can also suppress radiation loss if interposed between the absorber plate and the lower glass cover. Convection loss suppression also can be achieved, leading to improvement of overall efficiency. Low- to moderate-priced plastic film does not appear to have sufficient resistance to damage by high collector plate temperatures, however, so this technique has not been commercially utilized.

The foregoing discussion has been concerned with methods for reducing U_L, the heat loss coefficient, to the lowest practical level. By so doing, the total heat loss is minimized and collector efficiency is increased. It is evident from Equation (11-1) that losses also decrease as the difference between plate temperature and air temperature decreases. The ambient (outside) air temperature is an uncontrollable factor, of course, but the fact that it varies with time and with geographic location means that collector efficiency will also be dependent upon these factors. It is clear, also, that a collector will be more efficient at lower plate temperatures than at high temperatures. But plate temperature is dependent largely on the way the collector is operated, that is, by the temperature of the fluid being circulated in contact with the plate, the rate of fluid circulation, and the type of fluid. Fluid temperature depends on conditions elsewhere in the system, whereas the other factors depend on the collector design and the operating conditions.

SOLAR ENERGY ABSORPTION

In Equation (11-1), the first term is the solar energy absorbed in the absorbing surface, which depends upon the solar energy incident on the tilted surface of the collector and is affected by collector orientation, as outlined in Module 4. This climatic variable can be measured or derived from tables of averages, and if not already converted, can be calculated for the proper collector position.

The transmissivity of the glass, τ, is a function of the quality of the glass and the angle at which the solar radiation reaches the glass. At normal incidence (solar beam perpendicular to the glass surface), one sheet of ordinary window glass reflects about 8 percent of the solar radiation. Two sheets of glass with air space between reflect about 15 percent. Impurities in the glass, principally iron, result in some radiation absorption; typical glasses 1/8 inch in thickness absorb one to five percent per sheet. Glass with reasonably low iron content may absorb about 2 percent per sheet, so at normal incidence, the total transmission of 2 sheets of glass can be approximately 80 percent. The value of τ is, therefore, 0.8.

Because the beam radiation from the sun strikes the collector at an angle which varies throughout the day, as well as seasonally, a weighted mean transmissivity is somewhat lower than this normal-incidence value. Precise calculations can be made, but a satisfactory approximation for a single-glazed collector can be based on a 10-percent average reflection loss and a suitable absorption loss dependent on glass quality. Assuming 2-percent absorption, an average transmissivity, τ, could be about 0.88. In a double-glazed collector, an effective transmission coefficient of 0.78 could be used with good quality glass.

If plastics are used for the transparent surfaces, transmission coefficients could be appreciably different, depending upon the characteristics of the plastics. Some have transmissivities moderately higher than glass, whereas others show lower values.

Methods for reducing the reflectivity of glass surfaces have been developed. Metallic films formed by vapor deposition are commonly used as lens coatings in photographic equipment. These interference layers are too costly for use in solar collectors. Another process involves a delicate etching of the glass surface by acid treatment, producing essentially a slightly porous silica surface. Solar reflectivities as low as 1 to 2 percent can be obtained under carefully controlled conditions. Total transmissivity of a double-glazed collector can thereby be increased to values above 90 percent. The cost-effectiveness of this substantial improvement in performance has yet to be established.

The solar absorptivity of the radiation-receiving surface, α, is dependent on the optical property of the materials exposed to solar radiation. Surfaces which appear black to the eye have high absorptivity for the visible portion of the solar spectrum, and usually also are good absorbers for the infrared portion of the solar radiation. Carbon black, numerous metal oxides, and most black paints have absorptivities above 0.95, that is, they absorb 95 percent of the solar radiation reaching the surface. The remainder of the solar radiation is reflected upwards through the glazing. The overall efficiency of the collector is strongly dependent on the absorptivity of this surface.

The most common types of absorber surfaces are heat-resistant black paints, usually applied by spraying, followed by curing with heat to eliminate solvents and to secure permanence. These surfaces must be capable of prolonged exposure to temperatures of 300 to 400 °F in double-glazed collectors, without appreciable deterioration or outgassing. In a recently developed solar air collector, sheet steel coated with black porcelain enamel (applied to the steel as a sprayed-on frit and fused to the surface in a furnace) is achieving successful application.

SELECTIVE SURFACES

Most surfaces that are good absorbers for solar radiation are also good radiators of heat. If, for example, a surface has an absorptivity of 0.95 for solar radiation, it will normally radiate heat at a rate about 95 percent of that of a "perfect" radiator. Certain combinations of surfaces, however, are capable of absorbing solar radiation effectively, while at the same time radiating heat at a low rate. These combinations are known as selective surfaces. Most selective surfaces are composed of a very thin black metallic oxide on a bright metal base. The black oxide coating is thick enough to act as a good solar absorber, with an absorptivity as high as 0.95; but it is essentially transparent to long-wave thermal radiation emitted by an object at a temperature of several hundred degrees F. Since bright metals have low emissivity for thermal radiation, that is, are poor heat radiators, and since the thin oxide coating is transparent to such radiation, the combination is a poor heat radiator. As a result, the radiation loss from this type of surface is considerably lower than from a conventional, non-selective surface. Thus, the overall heat loss coefficient, U_L, has a lower value when this type surface is used.

The most successful and stable selective surface developed to date is made by electroplating a layer of nickel on the absorber plate, then electrodepositing an extremely thin layer of chromium oxide on the nickel substrate. Nickel oxide coatings have also been used, but they are less resistant to damage from moisture. Coatings of copper oxide on bright copper and nickel have similar properties, but temperature stability is limited. The most effective selective surfaces have solar absorptivities near 0.95 and thermal emissivities near 0.1.

COLLECTOR PERFORMANCE

CONVENIENT PERFORMANCE EQUATION

Having now recognized the principal design factors affecting collector performance, specifically those related to heat loss control and those involving the absorption of solar radiation, we now can see from Equation (11-1) that if the numerical values of all the terms are known, the rate of useful heat recovery, Q_u, can be calculated. In addition to the design characteristics of the collector discussed above, the three operation conditions, solar radiation, average absorber-plate temperature, and ambient temperature, must be known. With the exception of plate temperature, these terms can readily be measured or obtained from tables or charts. Absorber-plate temperature, however, is seldom known, nor can it be easily determined. It is affected by the other collector operating conditions and, most critically, by the temperature of the fluid being supplied to the collector to be heated.

In an operating system comprised of collector, storage, and space being heated, the temperature of the fluid in storage can be measured. When a system is being designed for a building, storage temperature can be calculated or assumed until confirmed. This fluid is supplied to the collector and strongly controls the absorber-plate temperature in Equation (11-1). In a typical liquid collector, average plate temperatures usually are 10 to 20 degrees above inlet liquid temperature, and in air collectors, the temperature difference is 30 to 50 degrees. As a convenience, therefore, Equation (11-1) can be modified by substituting inlet fluid temperature for the average plate temperature, if a correction factor is applied to the resulting useful heat determination. The resulting equation is

$$Q_u = F_R A_c [H_T \tau\alpha - U_L (T_i - T_a)] \qquad (11\text{-}2)$$

where

T_i is the temperature of the fluid entering the collector

F_R is a correction factor or "heat recovery factor", having a value between 0 and 1.0, such that the useful heat recovery calculated by Equation (11-2) is equal to that calculated by Equation (11-1).

HEAT RECOVERY FACTOR

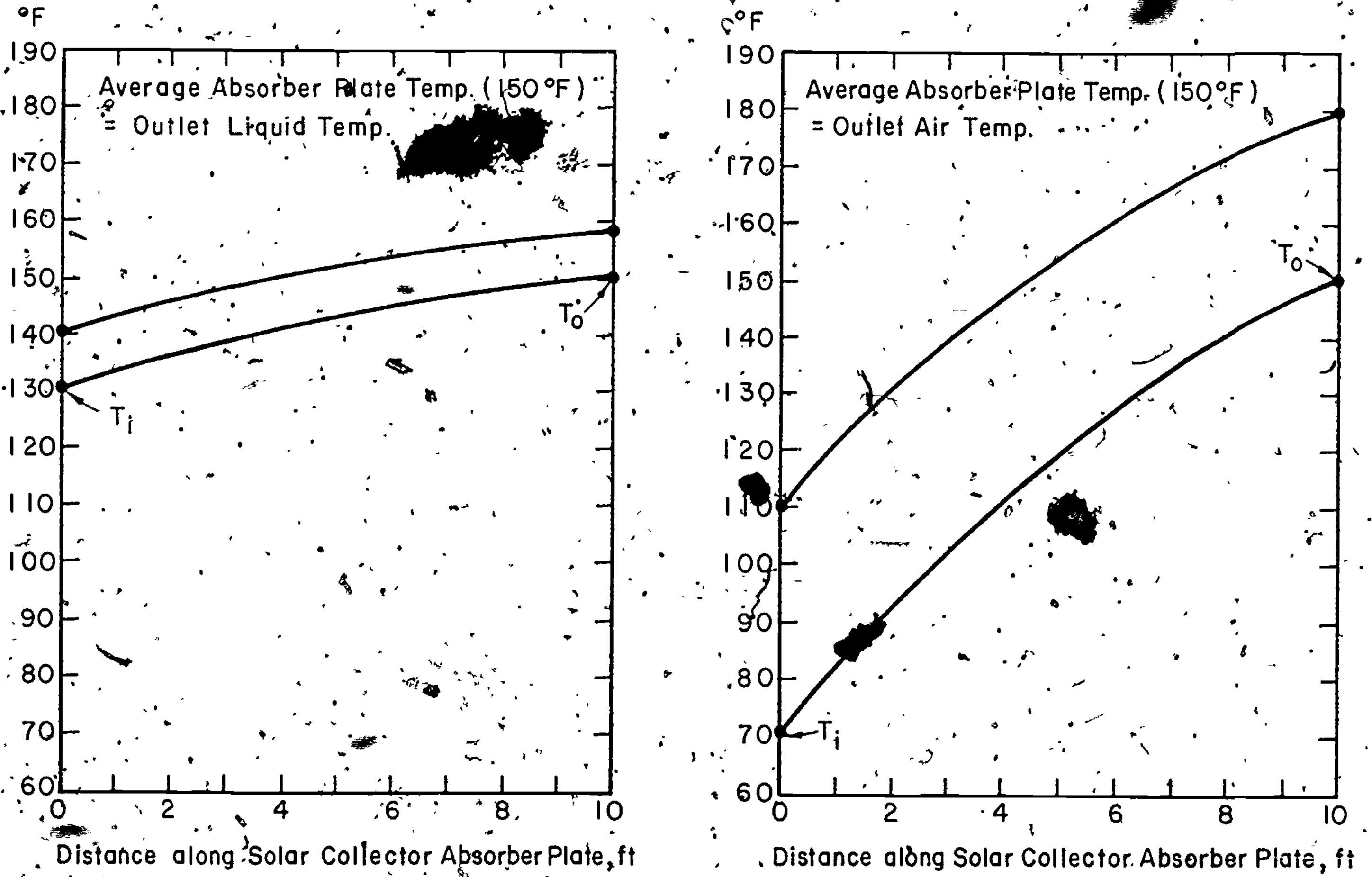
The heat recovery factor, F_R, can be interpreted as the ratio of the heat actually recovered to that which would be recovered if the collector plate were operating at a temperature equal to that of the entering fluid. This temperature equality would theoretically be possible if the fluid were circulated at such a high rate through the collector that there would be a negligible rise in the temperature of the fluid passing through the collector, and the heat transfer coefficient were so high that the temperature difference between the absorber surface and the fluid would be negligible.

In Equation (11-2), the temperature of the inlet fluid is dependent on the characteristics of the complete solar heating system and the heat demand of the building. F_R, however, is affected only by the collector characteristics and the fluid flow rate through the collector. As indicated above, the numerical value of F_R would be 1.0 if the entering fluid temperature and the average plate temperature were the same.

COLLECTOR TEMPERATURE PATTERNS

The better the heat transfer coefficient between the metal plate and the fluid, the more nearly the fluid temperature will approach the plate temperature at any one position in the collector, hence the higher will be the value of F_R. Similarly, the greater the fluid circulation rate, the smaller will be the temperature change from inlet to outlet and the closer will be the inlet fluid temperature to the average plate temperature. Figure 11-4 shows a typical temperature pattern in a solar heater being supplied with liquid at 130 degrees. Liquid leaves the collector at about 150 degrees, the collector-plate temperature is about 10 degrees above the liquid temperature throughout the collector, and the average plate temperature is about 150 degrees. If typical values of the collector parameters are substituted in Equations (11-1) and (11-2), it will be found that using 130 ^{0}F as inlet fluid temperature in Equation (11-2) instead of 150 ^{0}F as the average plate temperature in Equation (11-1) would necessitate use of a heat recovery factor, F_R, of about 0.9 to obtain the correct value of Q_u. If the coefficient of heat transfer between the collector plate and the liquid is lower, or if a lower fluid circulation rate is used, the value of F_R would be slightly less.

A temperature pattern in a typical air-heating collector operating with an air supply from the space being heated or from the cold end of a pebble-bed storage unit at 70 ^{0}F is also shown in Figure 11-4. Full sun and a practical air circulation rate of about 2 cfm per square foot of collector are assumed in the example. An air temperature rise of about 60 to 80 degrees would occur under these conditions, which is much higher than in the liquid case because of the lower specific heat for air. The mass flow rate is about the same as that of the liquid (measured as pounds per hour, for example) for suitable pressure loss conditions. Rather than a moderate



Liquid Heating Solar Collector

Air Heating Solar Collector

Figure 11-4. Comparison of Typical Temperatures in Liquid and Air Heating Solar Collectors.

10-degree difference between plate and fluid temperatures, as in the liquid case, the air collector is characterized by a 30-to-50-°F temperature driving force. The much lower heat transfer coefficient from the plate to the air is responsible for this difference. Under the conditions chosen, the average plate temperature would be about 150 degrees, approximately the same as that estimated for the liquid system. Use of Equation (11-2) with an inlet temperature of 70 degrees results in a heat recovery factor, F_R, typically about 0.7 for the air collector. Characteristically, solar air heaters having heat transfer surfaces approximately equal to the solar absorbing area show heat recovery factors substantially below those achieved in liquid collectors. However, as shown below, this difference must not be interpreted as superiority of one over the other when used in suitably designed systems.

COLLECTOR EFFICIENCY

Equation (11-2) may be rewritten as an efficiency of solar collection, that is, the ratio of useful heat delivery divided by the total solar radiation, by dividing both sides of the equation by H_T and by A_c. Equation (11-3) is the result.

$$\frac{Q_u}{H_T A_c} = F_R \tau\alpha - F_R U_L \frac{(T_i - T_a)}{H_T} = \text{collector efficiency} \qquad (11\text{-}3)$$

For a given collector operating at a constant fluid circulation rate, A_c, F_R, τ, α, and U_L are nearly constant regardless of solar and temperature conditions. Assuming that they are constant, Equation (11-3) represents a straight line on a graph of efficiency versus $\frac{T_i - T_a}{H_T}$. The characteristics of this line are an intercept (the intersection of the line with the vertical efficiency axis) equal to the numerical value of $F_R \tau\alpha$ and a "slope" of the line, that is, the vertical scale change divided by the horizontal scale change, equal to $(-F_R U_L)$. So if experimental data on collector heat delivery at various temperatures and solar conditions are plotted on a graph with efficiency as the vertical axis and $\frac{T_i - T_a}{H_T}$ as the horizontal axis, the best straight line through the data points is a complete representation of the collector performance over its entire operating range. Where the line intersects the vertical axis corresponds to the fluid inlet temperature being the same as the ambient temperature, and collector efficiency is at its maximum. Where the line intersects the horizontal axis, collection efficiency is zero. This situation corresponds to such a low radiation level or such a high temperature of the fluid supply to the collector that heat losses are equal to solar absorption and no useful heat is delivered from the collector.

Typical Collector Characteristics

Figure 11-5 shows efficiencies of several types of collectors correlated in this way. These lines are the results of actual measurements. Collectors 4 and 7 are seen to have the highest efficiencies, but final selection also depends on costs, durability, appearance, and so on. Collector 4 appears to have the best performance of all those compared in Figure 11-5 if normally operated at conditions represented by the left-hand side of the graph. Such conditions are low operating temperatures or high solar radiation. Near the right-hand side of the graph, however, collector 7 is more efficient than collector 4, where high inlet collector temperatures or low solar radiation prevail. It is evident that some collectors are better than others in some temperature and radiation ranges, whereas a reversal can occur at different conditions.

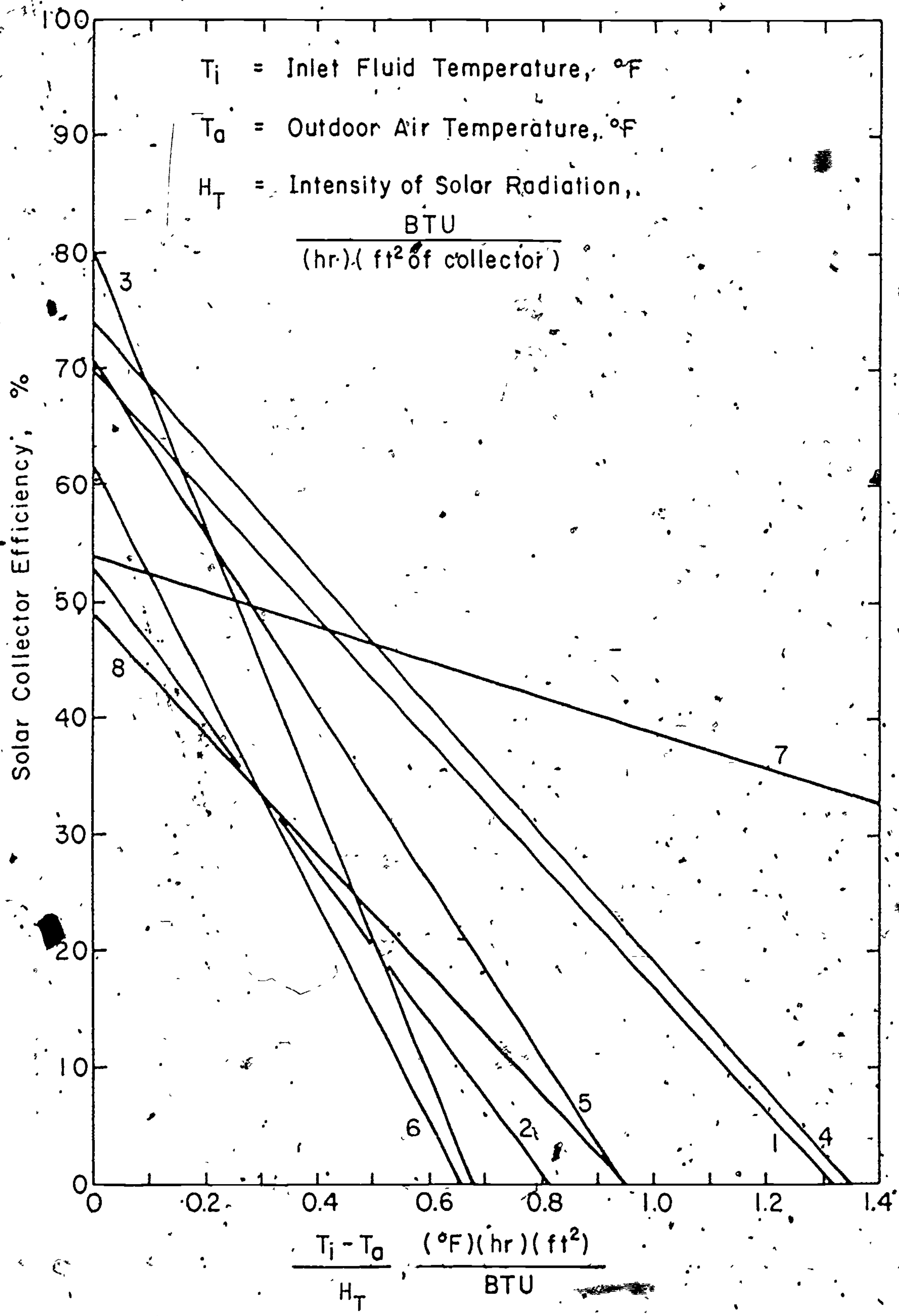


Figure 11-5. Solar Collector Efficiency.

A graph such as that in Figure 11-5 for a particular collector, when compared with others of similar type, can be used for selecting suitable equipment. Collector manufacturers usually provide such data. Of equal value are dependable data on the quantities $F_R\tau\alpha$ and F_RU_L. Knowledge of those two factors is equivalent to having the graphical relationship. Table 11-1 contains this information for the same collectors shown in Figure 11-5.

Comparison of Liquid and Air Collector Performance

Figure 11-6 shows efficiency relationships for a widely used air collector operating at two different air circulation rates, and a liquid collector (collector 5 from Figure 11-5). Whereas flow rate does not significantly affect the efficiency of a liquid collector, it is evident that air flow rate has a significant influence on air collector performance. Although even greater efficiencies can be achieved with higher air flow rates, the larger pressure drop and power requirements to circulate air at rates above 2 cfm/ft^2 force a compromise between collector efficiency and power consumption. Figure 11-6 also shows that at the same inlet temperature, ambient temperature, and solar radiation level, the liquid collector is more efficient. It is important to recognize, however, that liquid and air collectors normally operate at very different inlet temperatures, so that air collectors usually operate at conditions substantially nearer the left side of the graph than do the liquid type. The net result is comparable operating efficiency with the two types.

Table 11-1. Solar Collector Performance Parameters

Collector Number	Manufacturer and Remarks	Absorber Material	Absorber Surface Coating	Transparent Covers	F_R	U_L, $\frac{\text{Btu}}{(\text{hr})(\text{ft}^2)(^\circ\text{F})}$	$\tau\alpha$	α	ϵ	τ
1	NASA/Honeywell	Aluminum	Black Nickel	2 Glass	0.94	0.56	0.74	0.95	0.07	0.78
2	MSFC	Aluminum	Black Nickel	2 Tedlar	0.95	0.69	0.56	0.73	0.1	0.77
3	NASA/Honeywell	Aluminum	Black Paint	1 Glass	0.90	1.3	0.89	0.97	0.97	0.92
4	NASA/Honeywell (mylar Honeycomb)	Aluminum	Black Paint	2 Glass	0.96	0.57	0.77	0.97		0.79
5	NASA Honeywell	Aluminum	Black Paint	2 Glass	0.93	0.80	0.76	0.97	0.97	0.78
6	PPG	Aluminum	Black Paint	2 Glass	0.85	1.1	0.73	0.95	0.95	0.77
7	Owens (evacuated tube)	Glass	Selective Surface	1 Glass	0.75	0.20	0.72	0.8	0.07	0.9
8	Solaron (data furnished by manufacturer). Heat transfer fluid is air.	Steel	Black Paint	2 Glass	0.67	0.77	0.73	---	---	---

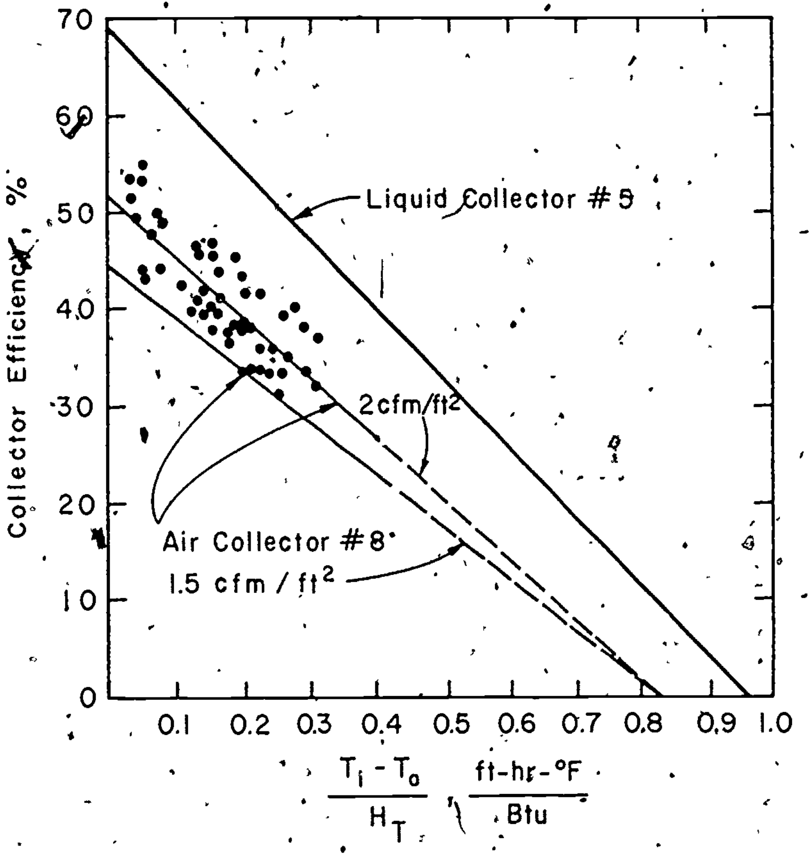


Figure 11-6. Comparison of Liquid and Air Collectors Based on Measured Performance (points shown are for air collector operating at 2 cfm/ft^2).

The foregoing comparison leads to the conclusion that whereas similar types of collectors such as flat-plate liquid heaters can be compared by means of a graph such as Figure 11-5, comparisons cannot be drawn in this way between different types. A second conclusion is that since the conditions under which the collector must operate depend on system conditions, particularly storage temperature, comparative evaluation requires attention to the other components in the system and their effect on collector performance.

Table 11-2 contains a step-by-step summary comparison of air and liquid types of collectors. Typical air and water heaters are compared at a high solar radiation level at a fairly low solar intensity. Characteristic designs and operating conditions have been assumed.

Figure 11-7 shows the results of the two calculations in graphical form. It may be noted that at the high solar radiation level, 300 $Btu/hr \cdot ft^2$, the two collectors have identical (50%) efficiency, and at the lower solar level, 150 $Btu/hr \cdot ft^2$, the air collector (operating at the characteristically low return air temperature) has an efficiency substantially greater than the liquid collector.

In another section of this manual, methods for appraising the performance of complete systems under varying atmospheric and solar conditions are presented.

COMPARISON OF LIQUID AND AIR-HEATING SOLAR COLLECTORS

Solar air collectors have not achieved the degree of use which liquid collectors have, perhaps because of the prior art in solar water heating in warm climates. Evacuated glass tube solar collectors, because of their high efficiency and relatively low material costs, offer the possibility

Table 11-2. Comparison of Typical Solar Heating Systems Employing Liquid and Air Collectors

Performance Relationship:

Collection Efficiency: $$\frac{Q_u}{A_c H_T} = F_R \tau\alpha - F_R U_L \left(\frac{T_i - T_a}{H_T}\right)$$

Design Characteristics:	Liquid	Air
Heat Recovery Factor F_R	0.9	0.7
Heat Loss Coefficient U_L	0.75	0.75
Cover Transmission τ	0.85	0.85
Plate Absorptivity α	0.95	0.95
$F_R \tau\alpha$	0.73	0.57
$F_R U_L$	0.68	0.53

Operating Conditions:				
Atmospheric Temperature T_A, °F	30	30	30	30
Fluid Inlet Temperature T_i, °F	130	130	70	70
Solar Radiation H_T, Btu/hr·ft^2	300	150	300	150
Fluid Flow Rate, gpm/ft^2, cfm/ft^2	0.02	0.02	2.0	2.0
$(T_i - T_a)/H_T$	0.333	0.666	0.133	0.266
Calculated Performance:				
$F_R U_L (T_i - T_a)/H_T$	0.23	0.46	0.07	0.14
$F_R \tau\alpha - F_R U_L (T_i - T_a)/H_T$	0.50	0.27	0.50	0.43
Collection Efficiency, %	50	27	50	43
Computed Outlet Temperature, °F	145	138	134	125

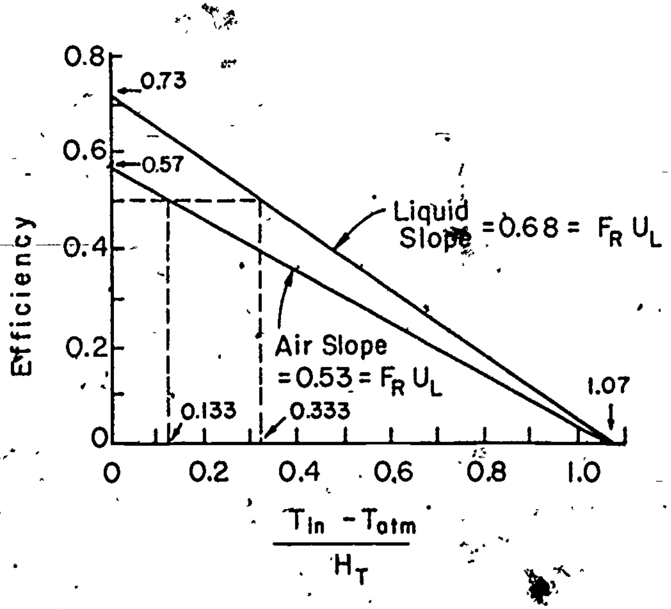


Figure 11-7. Results of Performance Calculations.

of substantially improving the overall performance of liquid systems. A double glass wall tube with the annular space between the two concentric glass tubes evacuated offers the possibility of some improvement, even for solar air heating collectors, if they can be produced and sold at competitive prices.

As to solar collector failure, the rate is much greater for liquid systems used for heating buildings. On the other hand, at least one manufacturer of air-heating solar collectors guarantees its product for 10 years.

Concentrating solar collectors for the heating of buildings do not appear to be practical in areas of the world where a large fraction of the total solar energy received is in the form of diffuse solar radiation. While flat-plate solar collectors collect diffuse solar radiation along with direct-beam solar radiation, concentrating solar collectors collect direct-beam solar radiation only.

In air systems, the problems of designing for solar collector overheating during periods of no energy removal are minimized. Warm-air heating systems are already in common use. Conventional control equipment is readily available for application to air systems.

Disadvantages of air systems include relatively high fluid circulation costs (especially if the rock heat-storage unit is not carefully designed), relatively large volumes of storage (roughly three times as much volumne as for water heat storage), a higher noise level, the difficulty of adding conventional absorption air conditioners to air systems, and the space required for ducting.

Advantages of air systems include no corrosion problems, no boiling problems, less concern for leakage, no freezing worries, and possibly lower maintenance costs than for liquid systems.

Advantages of water-heating solar systems include use of a common heat transfer and storage medium in areas of the world where freezing temperatures are not encountered. The water storage volume is about one-third of the volume of rocks for air systems to store equal quantities of heat. Liquid systems are rather easily adapted to supply energy to absorption air conditioners. They are also less noisy than air systems and are more readily adaptable to various architectural arrangements. The energy requirements for pumping the heat-transfer fluid range from 6 to 8 percent of the useful solar energy delivered.

In those areas of the world where freezing temperatures are encountered, the liquid used in the collector is separated from the water in the storage tank by means of a heat exchanger. This results in a temperature difference between the collector inlet temperature and the storage water temperature, hence in a higher absorber-plate temperature. An additional water pump is required when a heat exchanger is used. Solar water-heating systems usually operate at lower temperatures than conventional hot water systems and therefore require additional heat transfer surface area or fan-coil units to transfer heat into the building. Liquid-heating collectors may also operate at excessively high temperatures (especially in the spring and fall when both heating and cooling loads are least) and means must be provided to avoid boiling in the collector and rupturing the tubes in the collector because of increased pressures. When aqueous solutions are used as the liquid, care must be exercised to minimize corrosion problems, especially in the collector.

COLLECTOR ARRAYS

When arranging individual collector units into arrays, it is important to achieve equal flows through each collector. An array of liquid-heating solar collectors could be arranged as shown in Figure 11-8, with headers at the top and bottom of the array. Satisfactory flow distribution is realized if the headers are sized so that the head loss (pressure drop) from the bottom to top of each collector (column) is about 96 percent of the total head loss from A to B.

An arrangement for an array of air-heating collectors is shown in Figure 11-9. The main manifold ducts are sized in accordance with the volumetric air flow rates, and in the scheme shown, the manifolds within the collectors are used as "headers". The specific arrangement will, of course, depend upon the design of the collector.

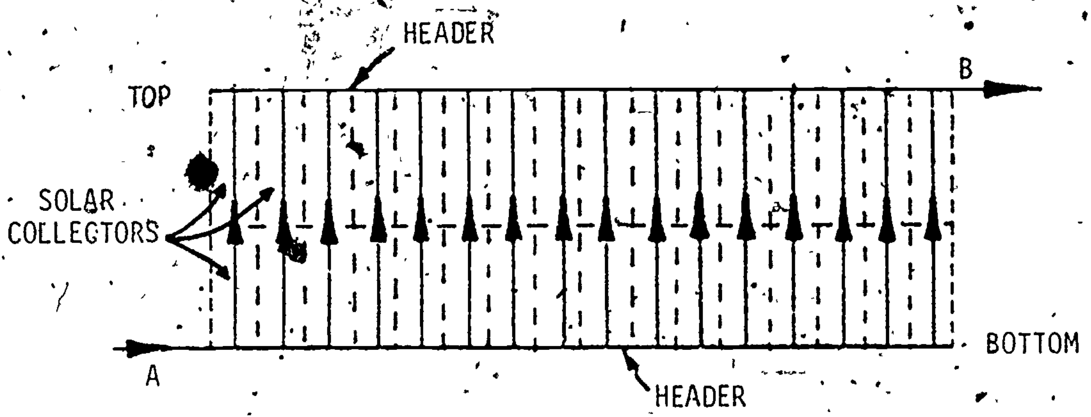

HEADER
TOP
B
SOLAR COLLECTORS
BOTTOM
A
HEADER

Figure 11-8. Definition Sketch for Fluid Flow Distribution, A Solar Collector Array.

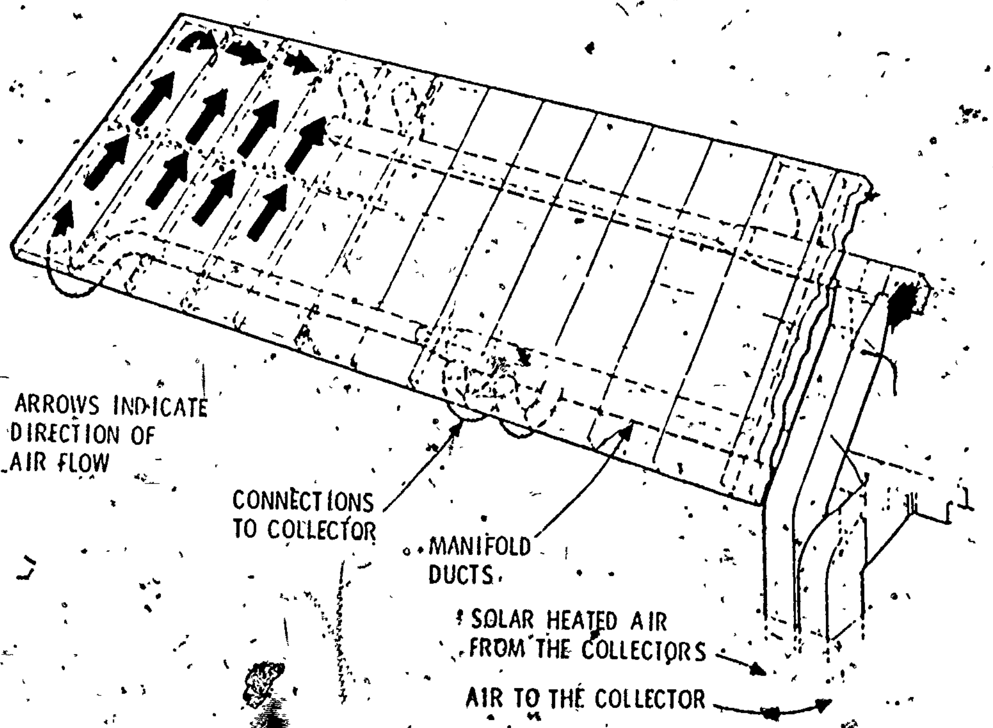

ARROWS INDICATE DIRECTION OF AIR FLOW
CONNECTIONS TO COLLECTOR
MANIFOLD DUCTS
SOLAR HEATED AIR FROM THE COLLECTORS
AIR TO THE COLLECTOR

Figure 11-9. Typical Arrangement of Internally Manifolded Collector Modules in an Array.

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TRAINING COURSE IN
THE PRACTICAL ASPECTS OF
DESIGN OF SOLAR HEATING AND COOLING SYSTEMS
FOR
RESIDENTIAL BUILDINGS

MODULE 12

STORAGE SYSTEMS

SOLAR ENERGY APPLICATIONS LABORATORY
COLORADO STATE UNIVERSITY
FORT COLLINS, COLORADO

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INTRODUCTION

TRAINEE-ORIENTED OBJECTIVE

The objective of this module is to distinguish and describe the design requirements for different types of heat storage for air-heating and liquid-heating solar systems.

SUB-OBJECTIVES

With the material in this module the trainee should be able to:

1. Identify the two principal types of heat storage used with solar systems
2. Describe the various aspects in the design and function of water storage, rock bed storage, and eutectic salt storage.

The purpose of thermal (or heat) storage in a solar heating and cooling system is to provide heat overnight and over intermittently cloudy periods during the day. The heat must be easily stored from the collectors, readily supplied to the heating and cooling system, have few internal losses or losses to the environment, be inexpensive and not take up an excessive amount of floor space. There are limitations to storage size for a given collector area. Several factors, the most important of which is cost, dictate that storage should be designed to serve an 18-to-30-hour time period.

TYPES OF STORAGE

The two principal types of storage, water storage and rock bed storage, are associated with a specific type of collection system.

Both types are based on sensible heat storage, in which the quantity of heat stored is directly proportional to the temperature rise in the storage medium. Storage of heat in water is generally used with a hydronic system and storage in a rock bed is generally used with a hot air system. Water has a high capacity for heat storage and, although rocks have one-fifth as much as water, both rocks and water are inexpensive.

Other types of heat storage material include metals. Iron, which is among the cheapest, is a material which has a high heat-storage capacity per unit volume. However, this material is about twenty times more expensive than rock for an equivalent heat storage capacity.

Another heat storage possibility is the use of phase change materials like eutectic salts (salt hydrates). These materials store latent as well as sensible heat. That is, they utilize the heat of liquefaction as the primary means of storing heat. Large amounts of heat can be stored and released by the process of melting and solidifying without change in temperature. The principal advantage in the use of these materials is that smaller storage size is needed as compared with a water tank or a rock bed. However, there are a number of problems that are associated with these materials that have been examined by researchers for years and have not as yet been resolved.

STORAGE CHARACTERISTICS

WATER STORAGE

Heat can be stored in a tank of water by circulating water from the tank through the collector loop and back to the tank either directly or by interfacing the tank and collector loop with a heat exchanger. Thus, the temperature of the entire tank is gradually increased. For non-pressurized tanks, the temperature will be limited to slightly below the boiling point of water. A non-pressurized tank should be vented, but the system size should be designed to prevent boiling. There is loss of energy associated with boiling. A pressurized tank is expensive and should not be considered for a normal residential heating and cooling system. Although there is a large amount of heat that can be stored as latent heat of vaporization, the high temperature at which this takes place is not needed in a solar heating and cooling system.

Water has a specific heat of one Btu per pound per degree Fahrenheit. On a volume basis, water can store one Btu/lb °F x 62.4 lb/ft^3 or 62.4 Btu per cubic foot degree Fahrenheit. The 1100-gallon or 147-cubic-foot water storage tank in CSU Solar House I can store about 9170 Btu per °F (147 ft^3 x Btu/ft^2 °F) of heat. Thus, if the tank is at 194°F and the tank is drawn down to 95°F (194°F - 95°F) (9170 Btu/°F), 908,000 Btu of energy would be provided. Solar House I has a heating load requirement of 17,600 Btu per degree Fahrenheit day. If the outside average temperature for one day is 14°F and the desired inside temperature is 68°F, then the load for a day would be 950,000 Btu (17,600 Btu/°Fday x (68°F - 14°F) x 1 day). Thus, if there is no loss of heat, the storage tank would have sufficient capacity to carry the building load for about 23 hours.

Water storage tanks should be insulated to prevent excessive heat losses. If the tank is located inside the building enclosure, the heat is not lost, but there is uncontrolled heat delivery to a localized region. In summer, the heat from the tank would add to the cooling load. If the tank is located underground, the heat is lost from the solar system.

The minimum useful temperature in a water storage tank for direct heating systems, as discussed in previous modules, is above 90°F.

For solar cooling systems, the minimum temperature is about 170°F for a lithium-bromide water absorption cooler. These lower useful temperatures and the boiling temperature as the upper limit set the temperature ranges which determine tank size in system designs.

In contrast to hot water storage tanks, the rock bed cannot be storing and delivering heat to the house at the same time. To heat the house from storage, the flow of air is reversed; cold air is delivered to the cold side of storage and is heated as it flows through the hot rock bed. Because the cold-air return to storage from the house is always at room temperature, the auxiliary heater can be placed in the delivery duct from storage. Auxiliary heat will not increase the temperature in storage. The temperature front in the rock bed will gradually recede, and even when all the useful heat has been depleted from the rock bed, the air can continue to be circulated through storage if the pressure drop across the bed is small.

ROCK BED STORAGE

Heat is stored in a rock bed by circulating heated air from the collectors directly through the rock bed. In contrast to the water storage tank, the rock bed is not heated uniformly, but is heated by layers to the temperature of the air stream coming from the collector. This results in temperature stratification, where the top of a rock bed is at the collector air temperature and the bottom of the rock bed is at room temperature. The advantages in stratification are that cold air is returned to the collector so that the collector operates more efficiently, and when heating from storage at night, the air temperature is high, being nearly at the same temperature as it was delivered from the collector during the day. In contrast, the limit of water storage temperature to heat the house is about 90°F, and the auxiliary boiler in a water system should not be placed in "series" with the storage tank because the water temperature return from the fan coil unit would be greater than 90°F and thus auxiliary fuel would be used to heat the storage tank.

Commonly available rocks have a specific heat of about .21 Btu/(lb)(°F). On a volume basis, the heat capacity is about 21 Btu/(ft^3)(°F) (.21 Btu/lb°F x 100 lb/ft^3) for .75-to-1.75-inch rock sizes. This is one-third the heat capacity of water on a volume basis. Thus, to have the same heat storage capacity as a water tank, about three times greater rock volume is required. The rock bed in CSU Solar House II contains 18 tons of .75-to-1.5-inch rocks in 363 cubic feet. At a uniform temperature of 150°F, 595,300 Btu (30 Btu/(ft^3)(°F) x 363 ft^3 x (150 - 68)°F) can be stored. Solar House II has a design heat load of 17,600 Btu/Degree Day. For an average outdoor temperature of 14°F, there would be approximately 17 hours of heating capability from storage.

PHASE CHANGE STORAGE

A phase change material, such as sodium sulphate hydrate, with its phase change occurring at about 88°F, could store a large amount of heat in a small amount of mass. This particular phase change material has a heat of fusion of 108 Btu per pound. To store 908,000 Btu of heat, 8407 pounds of salt hydrate would be required. This compares with about 9100 pounds of water with sensible heat storage from 90°F to 190°F. Because the density of the sodium sulphate hydrate is 91 lb/ft^3 as compared with 62.4 lb/ft^3 for water, there would be about 65 percent of the space required for the salt as for water. When costs are considered, the advantage of smaller quantity is lost. Also, other problems exist for this particular heat storage medium and for other phase change materials such as packaging in non-corrosive containers and maintaining chemical and physical stability in the cycling from solid to liquid states. Storage for solar heating and cooling systems is not sufficiently greater than water or rocks to warrant the greater costs.

DESIGN ASPECTS OF STORAGE SYSTEMS

WATER STORAGE

Tank Material

Tank materials most likely to be used in water storage are steel, aluminum, concrete, and plastics. A number of different modular and built-in-place tanks are shown in Figures 12-1 through 12-6. Steel tanks should be lined with a material such as butyl rubber to prevent internal corrosion from the water. Any material used as a lining should have a long life since replacement may be quite difficult. The lining materials must also withstand high temperatures that occur in the storage tank. Concrete tanks may not require lining, depending on quality, but require additional reinforcing and sealing of joints because of temperature stresses. Most plastic tanks currently available will not withstand the temperatures needed in solar heating and cooling systems. Therefore, special composition tanks will be required.

Tank Shape

A spherical tank provides the least surface area per unit volume of storage. It is cheapest to insulate. A shape that deviates from this, such as a long slender cylinder, requires more tank insulation material.

A spherical tank is also structurally advantageous. However, fabrication and support of spherical storage tanks are more difficult than for

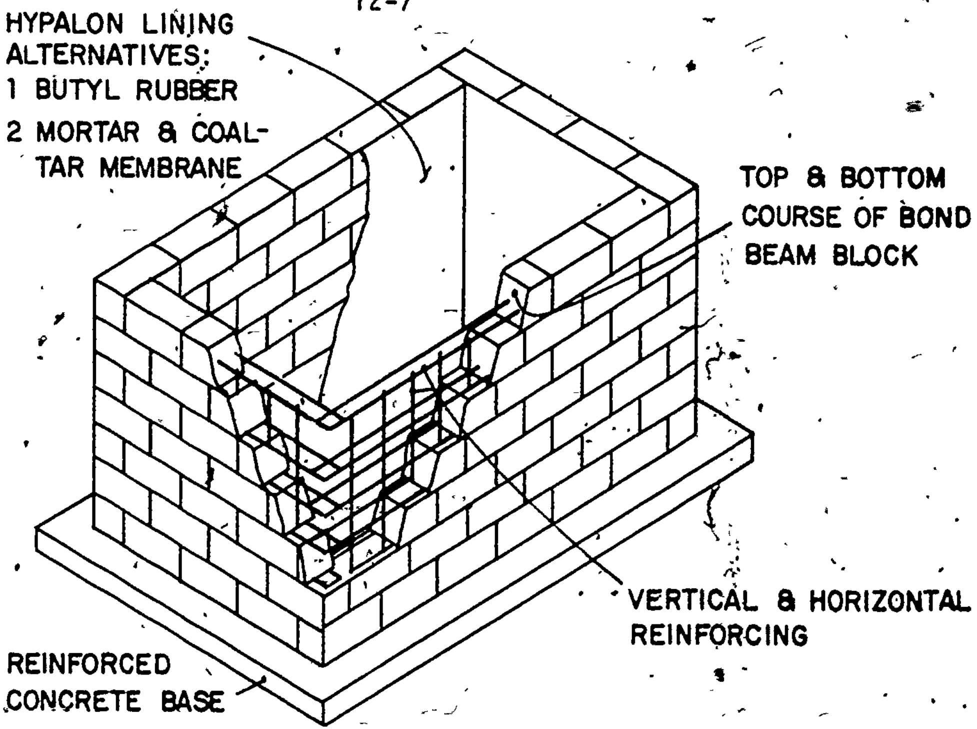

HYPALON LINING
ALTERNATIVES:
1 BUTYL RUBBER
2 MORTAR & COAL-TAR MEMBRANE
TOP & BOTTOM COURSE OF BOND BEAM BLOCK
VERTICAL & HORIZONTAL REINFORCING
REINFORCED CONCRETE BASE

Figure 12-1. Reinforced Concrete Block Tank

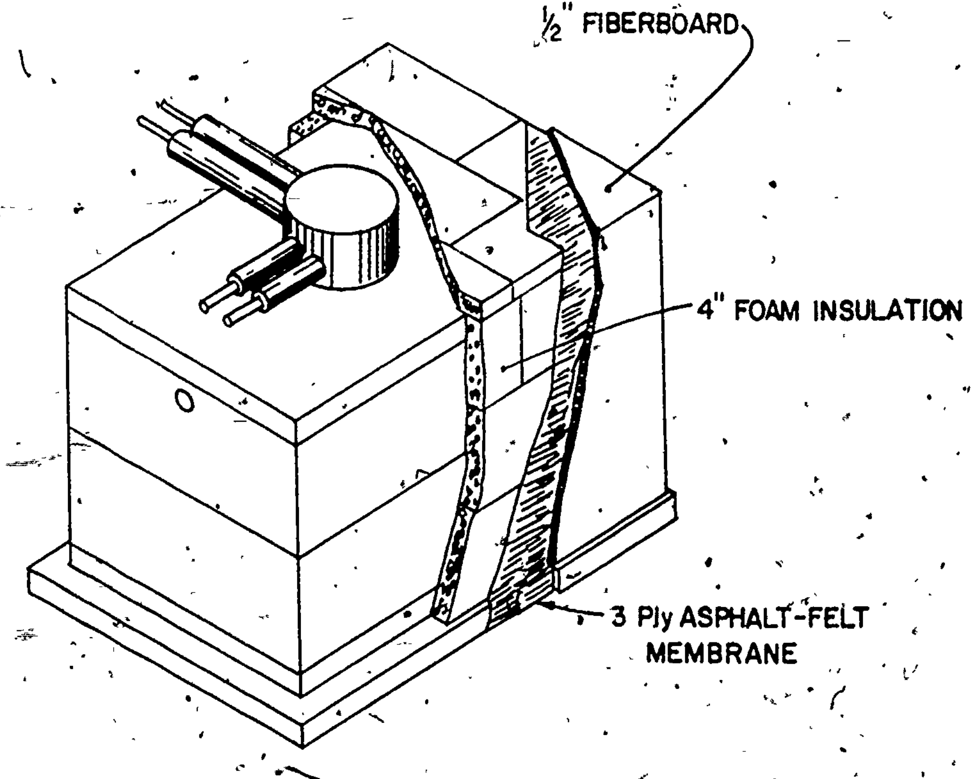

½" FIBERBOARD
4" FOAM INSULATION
3 Ply ASPHALT-FELT MEMBRANE

Figure 12-2. Precast Sectional Utility Vault

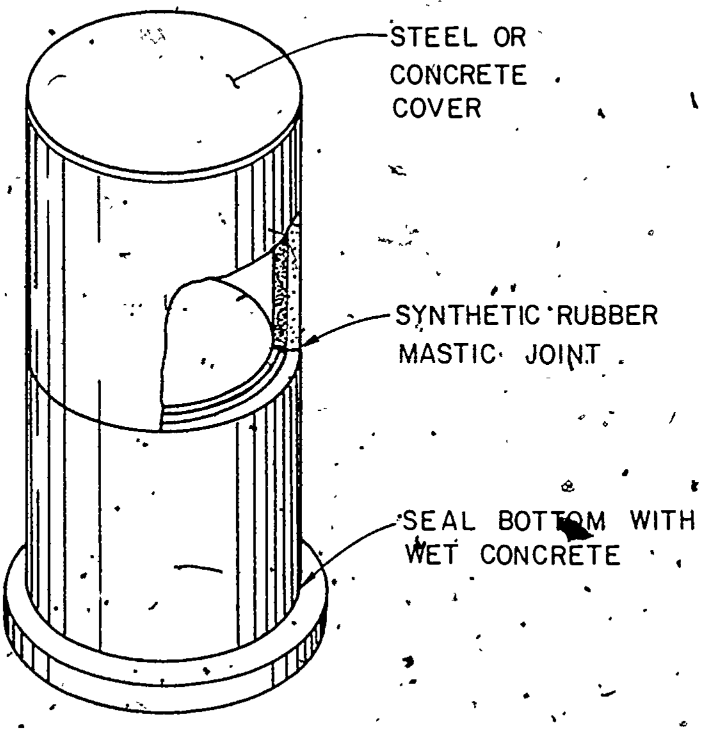
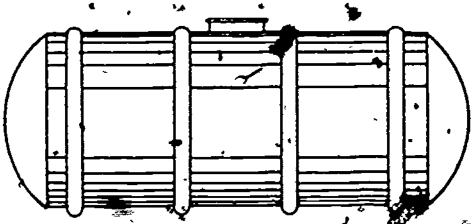
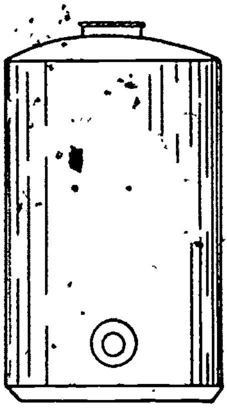

STEEL OR
CONCRETE
COVER
SYNTHETIC RUBBER
MASTIC JOINT
SEAL BOTTOM WITH
WET CONCRETE

Figure 12-3. Precast Concrete Storm Drain Pipe Tank



UNDERGROUND STORAGE TANK



ABOVE GROUND STORAGE TANK

Figure 12-4. Fiberglass Tanks

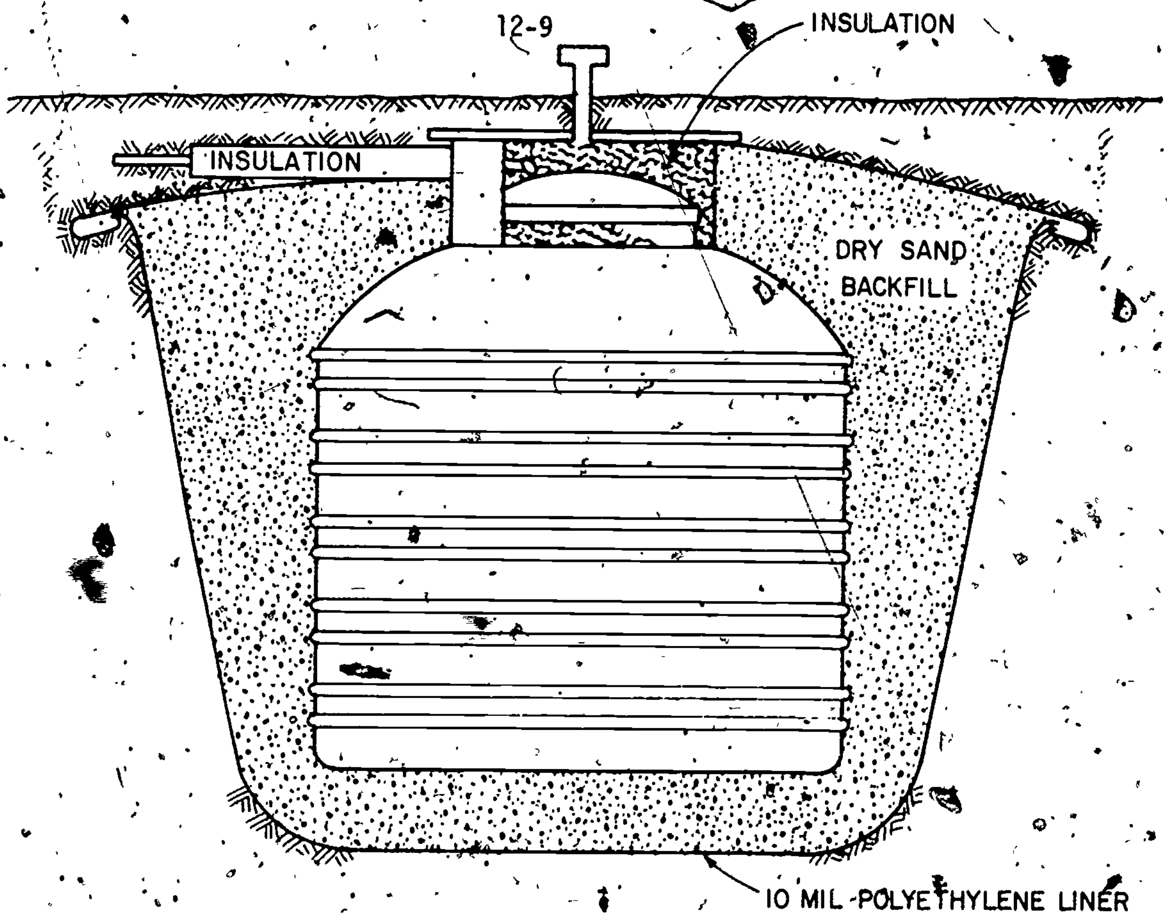

INSULATION
INSULATION
DRY SAND BACKFILL
10 MIL POLYETHYLENE LINER

Figure 12-5. Fiberglass Septic Tanks - Various Shapes - 500-1000 gal.

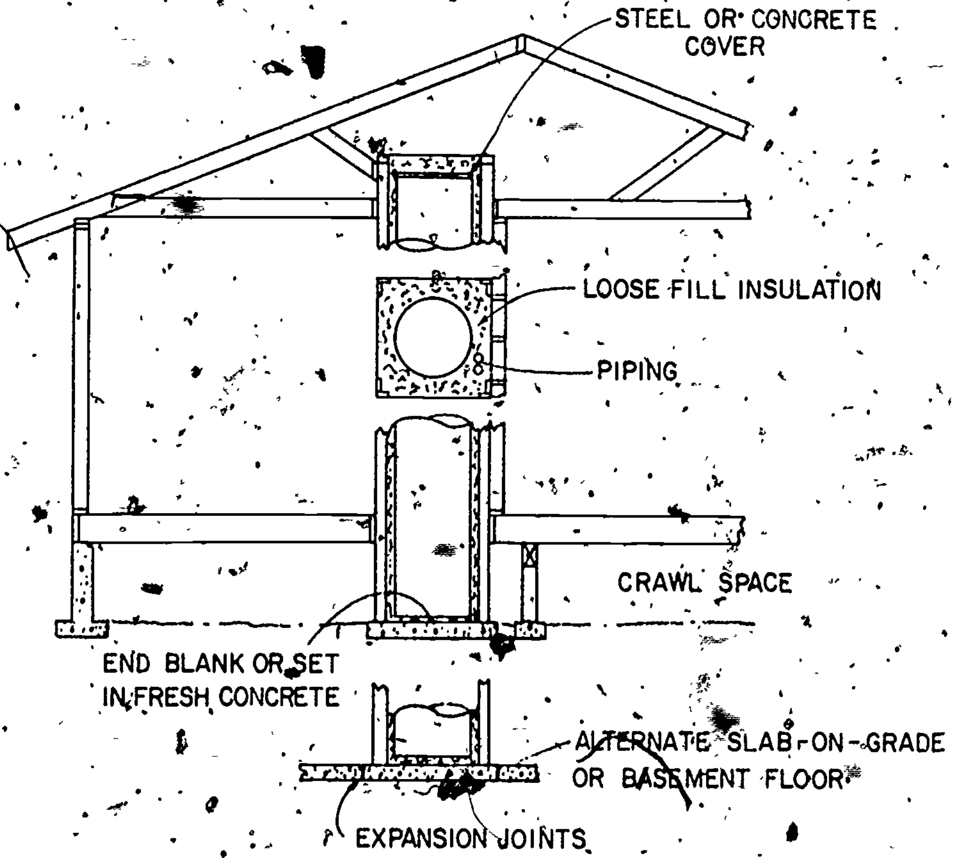

STEEL OR CONCRETE COVER
LOOSE FILL INSULATION
PIPING
CRAWL SPACE
END BLANK OR SET IN FRESH CONCRETE
ALTERNATE SLAB-ON-GRADE OR BASEMENT FLOOR
EXPANSION JOINTS

Figure 12-6. Vertical Tanks from Pipe Sections

cylindrical tanks. Thus, the most practical shape is a cylindrical tank having a diameter-to-width ratio of nearly one. Precast concrete tanks could be cylindrical or rectangular. Built-in-place reinforced concrete tanks would likely be rectangular because forming is easier.

Tank Insulation

Insulation of a storage tank is important to conserve the collected heat. The bottom as well as the sides of the tank should be insulated. Tank bottom insulation must be accomplished prior to installation of the tank. There are several possibilities for doing this. One is to rest the tank on rigid insulation foam pads. Another is to rest the tank on closely spaced two-by-six-inch boards and insulate between them.

Several different approaches can be used to insulate. One is to wrap the tank in conventional insulating material and another is to use a spray foam type of insulation. Insulation of at least R-23 is recommended for inside and R-30 for outside tank placements. A type of insulation that will not absorb moisture should be selected in outside locations or other areas where water or moisture may be a problem.

It is also desirable to enclose the tank along with the associated components such as the hot water heater and heat exchangers in a vented, insulated room. This isolates the tank and other heat-producing subsystems from the rest of the house.

Tank Location

If the tank is located within the building, there is some loss of living area. Usually the most desirable location for the tank is in the basement. In this case the tank is easily accessible for repairs. In other cases, such as retrofit applications or where there is no basement, other locations must be found. Alternatives are the garage or outside the house, either above ground or buried.

Tank Stratification

There is little stratification in a hot water storage tank. The difference in temperature of the water between the top and bottom of the tank is about 5°F during normal operation in 1000-gallon tanks. Stratification can be enhanced to some degree by the introduction of baffles to prevent convection and mixing. However, it is questionable that the gain is worth the expense. Another possibility is to use multiple tanks, but this adds to the expense of the system.

Two major advantages can be gained by stratification. One is higher temperature water delivered to the heating coils, and the other is colder water delivered to the collectors. Higher temperatures to the fan coils could result in smaller sizes and colder temperatures to the collector, resulting in increased collection efficiencies.

Piping to the Tanks

The inlet pipe to the storage tank from the collector should be located toward the top of the tank, and the outlet to the collector should be at the bottom of the tank. The outlet from the tank for house heating and cooling should be toward the top of the tank, where the tank is the hottest, and the return should be toward the bottom of the tank. Vented tanks should be provided with a make-up water line leading to the bottom of the tank with a float control valve.

Storage Tank Size

Studies have shown that for most locations in the United States, the storage tank should be sized to hold from 1.5 to 2.5 gallons of water per square foot of collector area. A small storage tank will have higher average temperatures and hence greater heat losses. However, high storage temperatures are desirable for air-conditioning applications, since the cut-off temperature for an absorption air-conditioning unit is about 170°F. A large storage tank will have lower average temperature and may not be able to provide direct heating of the house. For most residential applications and locations, the system performance is relatively insensitive within the 1.5 to 2.5 gallon per square foot range.

ROCK BED STORAGE

Container Arrangement

A rock bed storage bin can be constructed with wood. Stud walls with one-half inch plywood on both sides and 3.5 inches of R-11 insulation placed between the studs form an adequate storage container. A plenum must be provided on the top and bottom to distribute air flow evenly over the container cross-section, as shown in Figure 12-7.

The bottom plenum is constructed by supporting expanded wire mesh on concrete blocks placed about 1.5 inches apart. The rocks are then placed on top of the wire mesh and the bin is filled to within a foot of the top of the rocks. The space at the top of the bin above the rocks forms the top plenum.

The bin should be sealed before the rocks are placed to prevent air leakage. This is accomplished by caulking the joints with an epoxy or other suitable heat-resistant compound. Butyl rubber gasket or other heat-resistant material can be used to form the seal for the top lid. Air temperatures are nominally 150°F, but higher temperatures of 180° to 190°F are sometimes reached. Thus, sealant materials should be selected to within the higher temperatures.

Container Size and Shape

The rock bed storage should be sized to provide 50 to 100 pounds of rock per square foot of collector. For normal rock densities and for

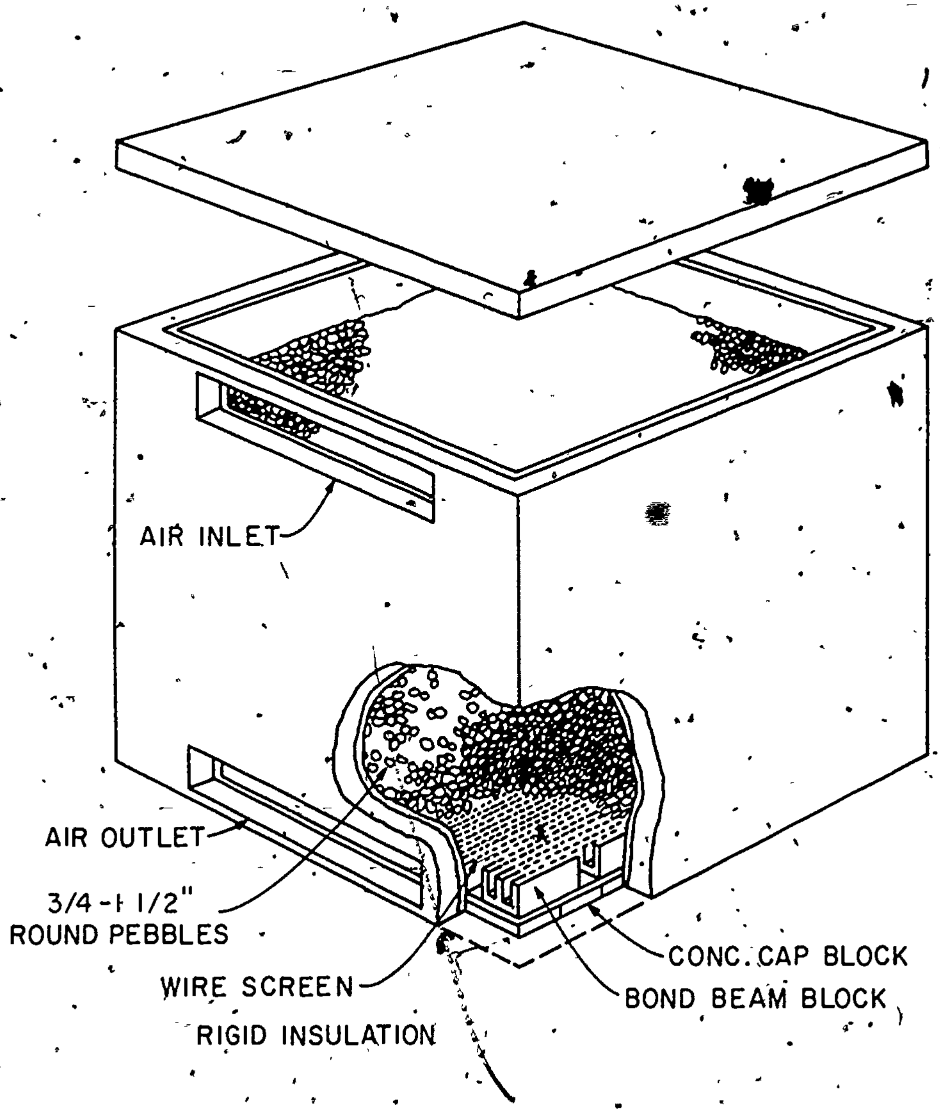

AIR INLET
AIR OUTLET
3/4-1 1/2"
ROUND PEBBLES
WIRE SCREEN
RIGID INSULATION
CONC. CAP BLOCK
BOND BEAM BLOCK

Figure 12-7. Rock Bed Heat Storage Unit

.75-to-1.5-inch sizes; this is equivalent to one-half to one cubic foot per square foot of collector.

Ideally a rock bed should have a short distance from inlet to outlet, with a large cross-sectional area perpendicular to the direction of air flow. With a large cross-sectional area, the air velocity is low and, coupled with a short travel path through the rock bed, the pressure drop is small. The smaller pressure drop results in lower fan power.

The rock bed must be deep enough to permit stratification. A minimum depth of 2.5 feet is recommended. In order to have adequate storage, however, the volume of the rock bed must be large. To avoid a large cross-sectional area with consequent displaced floor area, a larger depth may be used. When rock beds are constructed in the building, a depth of about five feet is allowable.

Figure 12-8 shows representative temperature profiles as they develop throughout the day for a typical 4.5-foot-high rock bed storage. In this figure, the bed is assumed to be fully discharged at the beginning of the day. From this figure it can be seen that a 2.5-foot-high bed will cause high outlet temperatures at the bottom after 1:00 p.m.

When storage is not fully discharged by morning, the temperature profiles of Figure 12-8 would be displaced to the right by the end of the day. A five-foot rock bed depth can therefore be advantageous.

Rock Size

The rock size is relatively unimportant, although it affects the pressure drop through the bed. The rock size should not be so small as to reduce flow rates significantly, nor so large that the interior of individual rocks is never heated. It is recommended that rocks from .75 to 1.5 inches in size be used.

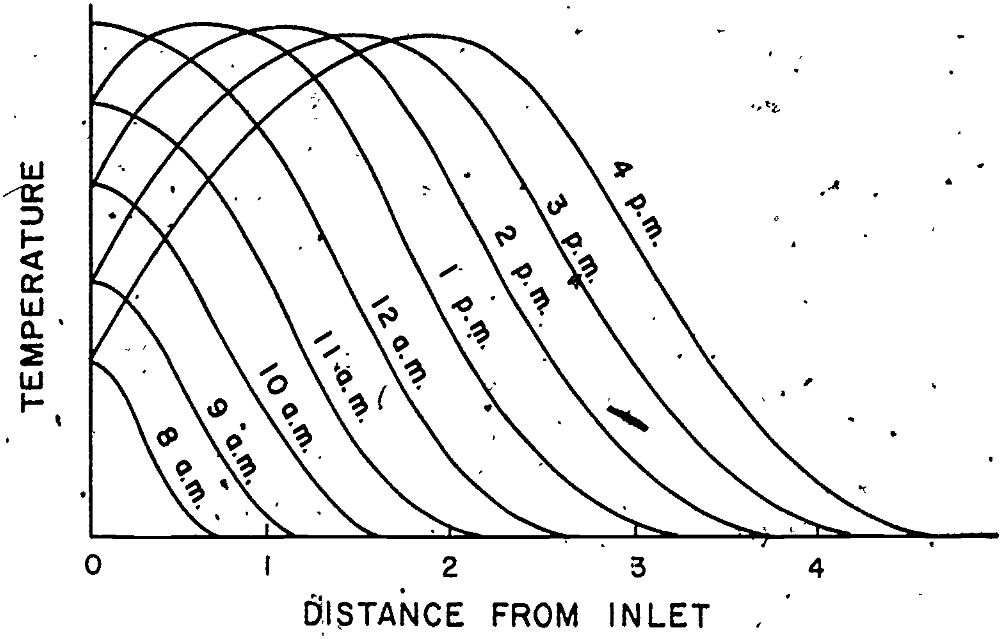


Figure 12-8. Typical Temperature Profiles in a Rock Bin Storage

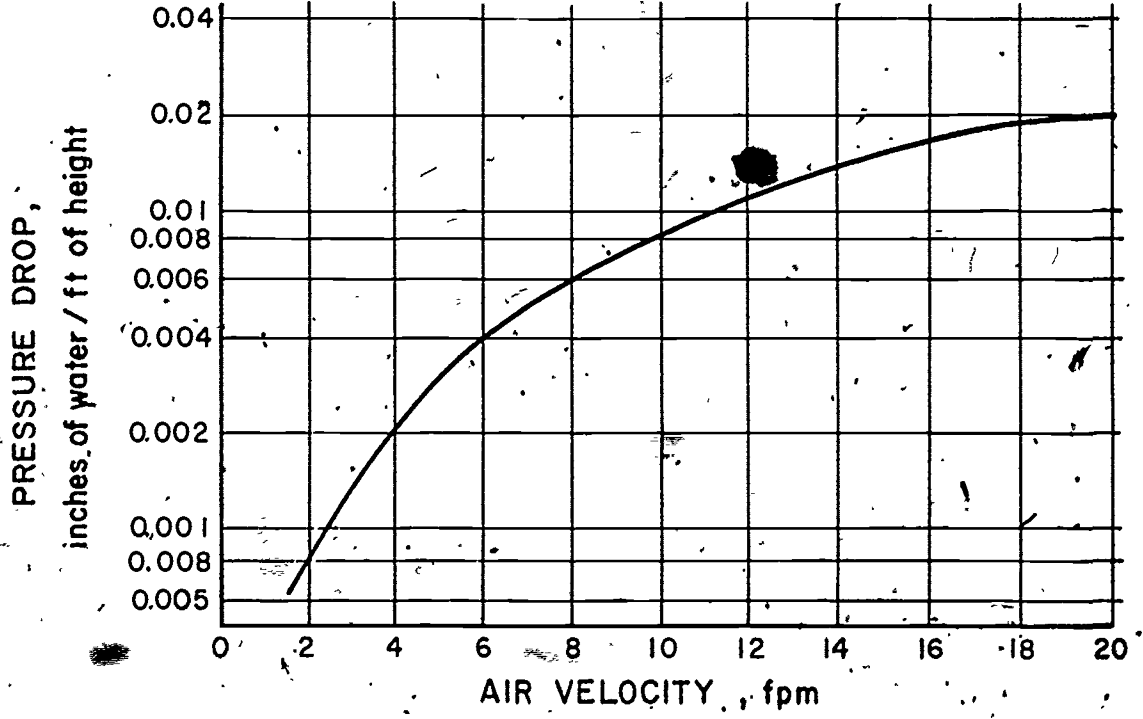


Figure 12-9. Pressure Drop Through a Rock Bin with 3/4 to 1-1/2 Inch Rocks

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Full Text Provided by ERIC

Rounded rocks are preferable to sharply fractured rocks. However, it is entirely satisfactory to use crushed gravel aggregates normally used for concrete.

Air Flow Rates

The flow rates through storage are governed by the pressure drop through the storage bed and the required flow rate through the collector to obtain the desired temperature rise. As noted in earlier modules, the desired flow rate is 2 cfm per square foot. Thus the storage cross-sectional area should be sized to keep the superficial velocity less than about 25 feet per minute. The superficial velocity is determined by dividing the flow rate by the cross-sectional area of the rock bed.

Figure 12-9 shows the pressure drop through a rock bed. Suppose the collector area for a system is 700 square feet. The flow rate is therefore 1400 cfm. Using a guideline for storage sizing of .75 cubic feet per square foot of collector area, the volume of the rock bed should be 525 cubic feet, or 26 tons of rock.

Air Flow Direction

The flow direction for the heat storage mode should be from the top to the bottom of the bed. When rock bed storage is placed in crawlspaces, it is necessary to direct the flow horizontally. The preferred flow direction is from top to bottom.

When heating the building from storage, the air flow should be reversed to take advantage of the stratification. If the rock pile is 4.5 feet high, then the cross-sectional area perpendicular to the air flow is 117 square feet, which is provided by a 10-foot by 11.7-foot (say 12-foot) size. The flow velocity is 12.0 fpm and the pressure drop is .05 in W.G. (from Figure 12-9).

Superficial air velocity less than 6 fpm should not be considered because heat transfer from the air to the rocks is poor and the cross-sectional area is unnecessarily large. Better economy is realized if cross-sectional areas are consistent with flow velocities through storage of about 15 to 25 feet per minute.

Fan Size and Power Requirements

Fan size and power requirements are governed by the pressure drops through the collector and distribution systems - when pressure drop through the rock bed is kept smaller than 0.1 in W.G. Sizing of blowers and motors is therefore determined by other components in the solar heating and cooling system.

Container Insulation

Insulation in the container walls need not be excessive. R-11 or R-19 insulation for the container should be adequate. Heat losses through container walls can be determined by the procedures outlined in Module 5. The overall U factor for wood container with R-11 insulation is about 0.07 Btu/(hr)(ft^2)(°F).

Special Precautions

The rocks should be washed before placement because excessive amounts of dust would clog the filter frequently. A potential problem could exist with cyclic heating and cooling of the rocks, which could cause some thermal fracturing.

With the evaporative cooling system, condensation on the rocks can be a problem during the summer. Condensation could cause mildew and odors.

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TRAINING COURSE IN
THE PRACTICAL ASPECTS OF
DESIGN OF SOLAR HEATING AND COOLING SYSTEMS
FOR
RESIDENTIAL BUILDINGS

MODULE 13

LABORATORY

SOLAR ENERGY APPLICATIONS LABORATORY
COLORADO STATE UNIVERSITY
FORT COLLINS, COLORADO

OBJECTIVE

At the end of this module the trainee should be able to make measurements of temperatures and flow rates at significant locations in a solar system to determine if the system is performing as desired.

INSTRUCTIONS

The class will form in two groups; one group will design solar systems and the other will work at the solar houses, and the groups will exchange assignments in the middle of the afternoon. The design group will refer to Module 14.

The group at the solar houses will further subdivide into four groups as follows:

Group 1: Start at Solar House I (entry)
Group 2: Start at Solar House II (conference room)
Group 3: Start at the air system model behind Solar House II
Group 4: Start at the water system model behind Solar House I

At the end of thirty minutes the groups should rotate, with Group 4 moving to the "top of the stack" and the other groups moving down one slot. This process is to be repeated at thirty-minute intervals until each group has been to all stations. The procedures to be followed at each station are described below:

STATION 1, SOLAR HOUSE I

The group will be given some performance data for some selected period. The data will indicate a problem. The group should identify, from the data, the source and nature of the problem and prescribe appropriate action to correct the problem.

STATION 2, SOLAR HOUSE II

Use same instructions as for Station 1.

STATION 3, AIR SYSTEM MODEL

The groups will calculate the efficiency of the collectors over a given time interval. Each group should determine what data are required, make the appropriate measurements, and then perform the required calculations.

STATION 4, WATER SYSTEM MODEL

Use same instructions as for Station 3.

TRAINING COURSE IN
THE PRACTICAL ASPECTS OF
DESIGN OF SOLAR HEATING AND COOLING SYSTEMS
FOR
RESIDENTIAL BUILDINGS

MODULE 14

COMPUTER-AIDED F-CHART CALCULATIONS

SOLAR ENERGY APPLICATIONS LABORATORY
COLORADO STATE UNIVERSITY
FORT COLLINS, COLORADO

INTRODUCTION

TRAINEE-ORIENTED OBJECTIVES

At the end of this module the trainee should be able to:

1. Operate the interactive F-CHART program.
2. Perform calculations relative to a solar system in, or near, his home location.

INSTRUCTIONS

The group that is on campus will take turns at operating the interactive F-CHART program. An effort will be made to have each person operate a computer and obtain results that are of specific interest to the trainee.

TRAINING COURSE IN
THE PRACTICAL ASPECTS OF
DESIGN OF SOLAR HEATING AND COOLING SYSTEMS
FOR
RESIDENTIAL BUILDINGS

MODULE 15

SYSTEM CONTROLS

SOLAR ENERGY APPLICATIONS LABORATORY
COLORADO STATE UNIVERSITY
FORT COLLINS, COLORADO

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INTRODUCTION

The only control for solar heating and cooling systems with which occupants need to be concerned is the thermostat in the building. However, there are many important controls needed for solar systems that automatically control the pumps and blowers, valves and dampers, and the auxiliary subsystems to collect and deliver the heat into storage and into the building. The design, function and strategy of the automatic system controls are discussed in this module.

TRAINEE OBJECTIVE

The objective of the trainee is to understand the function, mechanics, installation, and maintenance of control systems.

SUB-OBJECTIVES

At the end of this module the trainee should be able to:

1. Identify control functions
2. Understand control circuits
3. Recognize control methods and hardware
4. Specify control components
5. Install and maintain control systems.

CONTROL FUNCTIONS

IMPORTANCE OF CONTROLS

The basic function of the control system is to ensure that a maximum amount of energy will be collected from a solar system to provide the required heating or cooling load to the structure. Controllers are extremely important in a solar heating and/or cooling system. There is a tendency for a great deal of concern for the efficiency of a collector and less concern for effectiveness of the control system. The net result is reduced effectiveness of a system. For example, one might spend a great deal of money for a good selective surface on the absorber of a collector to realize an improvement in efficiency of approximately ten percent, whereas carefully designed controls could achieve that much or more for very little cost and effort.

COLLECTOR CONTROL STRATEGY

Most controllers at present are on/off controllers, that is, they command a pump to be either on or off, depending upon certain conditions. The typical control strategy with respect to controlling the flow of a transport medium in the collector loop is to start the collector pump whenever the collector fluid exit temperature is greater, by some set difference, than the tank temperature and to turn off the collector pump whenever the collector outlet temperature approaches the tank temperature. The temperature difference to start the flow is nominally set at 20 °F and to stop the flow, at 3 or 4 °F.

As a specific example, suppose that the storage temperature is 120 °F and the collector temperature is 50 °F when the sun rises. The collector temperature will gradually increase, and when it reaches 140 °F the controller will start the collector pump (assuming that the storage temperature is 120 °F). Then in the afternoon as the sun begins to set, the collector temperature will begin to decrease. Suppose that the storage temperature has reached 150 °F by 3:30 p.m. When the collector temperature decreases to 153 °F the controller will stop the collector pump. The temperatures sensed by the sensors depend upon their location in the system.

Ratio of Temperature Differences

If the temperature sensor for the collector is located at a point such that the sensor is rapidly cooled by the transport medium, the result can be that the collector pumps will cycle on and off repeatedly. This cycling can also occur if the difference between the temperature to start and to stop ~~the~~ system is not properly selected. The ratio between the on-to-off temperature differences should be approximately five to seven. In the example given in the preceding paragraph, the starting temperature difference was 20 °F and the stopping temperature difference was 3 °F. The ratio is slightly less than seven. A larger value for this ratio will reduce the total energy collected by the system, while a smaller value will cause cycling.

Freezing Protection

Some controllers are designed to incorporate an aquastat to compare temperature of the transport medium with some preset temperature such as the freezing temperature of water. Then, if the temperature of the fluid in the collector approaches this preset temperature, the pumps are automatically started to circulate the fluid or to heat the fluid from storage in order to prevent feeezing. This is not a recommended protection measure against freezing, because if there is a power failure during cold weather, the pumps will not operate and the collectors can freeze. It is preferred to use an antifreeze solution in the collector loop.

Two-Speed Pump

A two-speed pump may be considered as a possible way to regulate the temperature rise in the collector to improve collection efficiency. By changing to a slower flow rate during periods of low solar insolation, the system will collect heat at useful temperatures, whereas with a high flow rate, the temperature of the fluid at the collector outlet would be low and the control would stop the collector pump. When the solar radiation intensity is high, the flow rate can be increased. The fluid temperature would be reduced because of greater flow, and the collector will be more efficient.

CONTROL SYSTEM HARDWARE

The solar system controls consist of power relays which switch electric valves and pumps in the liquid system, or blowers and dampers in the air system, and auxiliary heating units in both systems, in response to temperatures or temperature differences. Controls for solar systems fundamentally serve the same functions as conventional HVAC controls; however, there are more control functions in solar systems, and also there are "interlocks" which prevent undesirable or hazardous sequences of operation.

A solar system supplier should provide the required control hardware or at least specify it, along with explicit wiring instructions. Building a control system at the site should be avoided unless experience in this practice is available.

THERMOSTAT

A two-stage heat, indoor thermostat is recommended for residential solar heating systems, and a two-stage heat, one-stage cool type is recommended for solar heating and cooling systems. Variations will feature "on", "off," or "automatic" fan control to circulate the room air, and "heat", "cool," or "automatic" switches from heating to cooling or vice-versa to meet the need.

When cooling is required, the single-stage cooling provides indoor space temperature control. There is a dead band, which is a small range in temperature between start and stop signals given to the controller which in turn controls the cooling system. The dead band for most thermostats is almost 5 °F. The heating operation is a bit more complex. Upon demand for heat, the first stage calls for the solar system to provide heat. If the building heat loss is greater than the solar system can provide, the temperature in the building will continue to drop to stage two and the auxiliary system will be called upon to provide heat. The auxiliary system can provide sufficient heat for the building by itself, or in combination with the solar system to raise the temperature in the room to the upper temperature limit of stage one which stops the heating system. The upper temperature dead band is nominally about 2 °F.

The thermostat is the only control with which the occupant needs to be concerned. Once the occupant sets the winter comfort control level to, say, 68 °F, and the summer comfort level to, say, 75 °F (or other suitable temperatures), he should not have to select or adjust any other control in the heating and/or cooling system.

TEMPERATURE SENSORS

Types

There are many types of temperature sensors that can be used in the control subsystem such as thermocouples, thermistors, silicon transistors, bimetallic elements and liquid or vapor expansion units. Bimetallic elements and liquid or vapor expansion units are seldom used because other temperature sensors are more durable and dependable. Thermocouples are frequently used for temperature measurement. However, they are not often used in controls because the voltage output is low, in the millivolt range, and without amplification the voltage is insufficient to be used in controls.

Thermistors and silicon transistors are used in the control subsystem because the voltage outputs from these sensors are in the 0-10 volt range and are high enough to serve the control functions. The voltage outputs from thermistors are non-linear, and calibration circuitry must be provided for the non-linearity. The voltage outputs from silicon transistors are linear in the normal operating temperature range of solar heating and cooling systems, and provide for simpler circuitry to control the system.

Location

The locations of temperature sensors are not particularly critical, but there are some preferred locations. Temperature sensors are required to measure the air or liquid temperature as it exists from the collector, in the solar storage tank, or rock bed, and in the preheat water tank. The sensor in the conditioned space is the thermostat.

The sensor which measures the fluid temperature at the collector outlet can be located in the manifold which collects the fluid from the total array of collectors. It is preferred that the sensor be in contact with the fluid, but it is acceptable for the sensor to be in contact with the pipe, provided there is good thermal contact of the sensor with the pipe. If the sensor is attached to the outside of the outlet pipe, the sensor should be well insulated so that it does not lose the heat to the surroundings and register a low temperature. It is important to locate the sensor near the outlet so that it can register the fluid temperature when the sun is heating the collector but the fluid is not circulating. Sensors in the outlet manifold will register the increase in temperature, but the sensor located far from the manifold will not, and useful energy cannot then be collected. Wherever the sensor is located, the characteristics should be checked out when the system is put into operation.

The sensor in the storage tank should be located near the bottom third inside the tank. When there is no fluid circulation, the temperature at the top of the tank will be slightly higher than the bottom, but while the fluid is in circulation, the fluid in the tank is usually well mixed and the temperature will be uniform.

The location of the sensor in the preheat tank should be near the top one-third of the tank. If it were located near the bottom, the temperature at the top could be several degrees hotter. Also, when hot water is used in the household, cold water enters the preheat tank near the bottom. While the preheat tank would be thermally mixed when the pump is started, frequent cycling could result from the sensor registering locally cold water temperature. For an air system, the cycling is not particularly harmful because only one pump for the preheat cycle is involved. However, for the hydronic system, two pumps will be put into operation, and frequent cycling can be wasteful of electric energy. In both air and liquid systems, more heat would be lost than necessary from the pipes and heat exchangers because of frequent cycling.

The sensor in the pebble bed should be located at the bottom (or outlet) end of storage. When heat is being stored, the bottom (or outlet) end of storage will determine if storage is "full".

CONTROL PANELS

Usually a central control panel is convenient to consolidate the circuits and relays that provide the control functions. The panel would house the relays and provide for some adjustment of the temperature limits. It is best to acquire a control panel from the solar equipment manufacturer as a prewired unit to serve the system. All that needs to be done with a prewired control panel is to connect the temperature sensors, and motor, auxiliary, and valve and damper controls to the proper terminals in the control panel. The manufacturer will provide the necessary hook-up instructions. The power for the control panel will usually be household 115-volt single-phase A.C. line power.

TYPICAL CONTROL SUBSYSTEMS

AIR SYSTEM

A sketch of a typical configuration for an air system is shown in Figure 15-1. The temperature sensors are indicated by T_{ci}, T_{co}, T_S and T_E. T_{ci} measures the collector inlet temperature, T_{co} measures the collector outlet temperature, T_S measures the storage temperature, and T_E measures the enclosure temperature. BD-1 and BD-2 represent backdraft dampers, D-1 and D-2 represent manual dampers, and MD-1 and

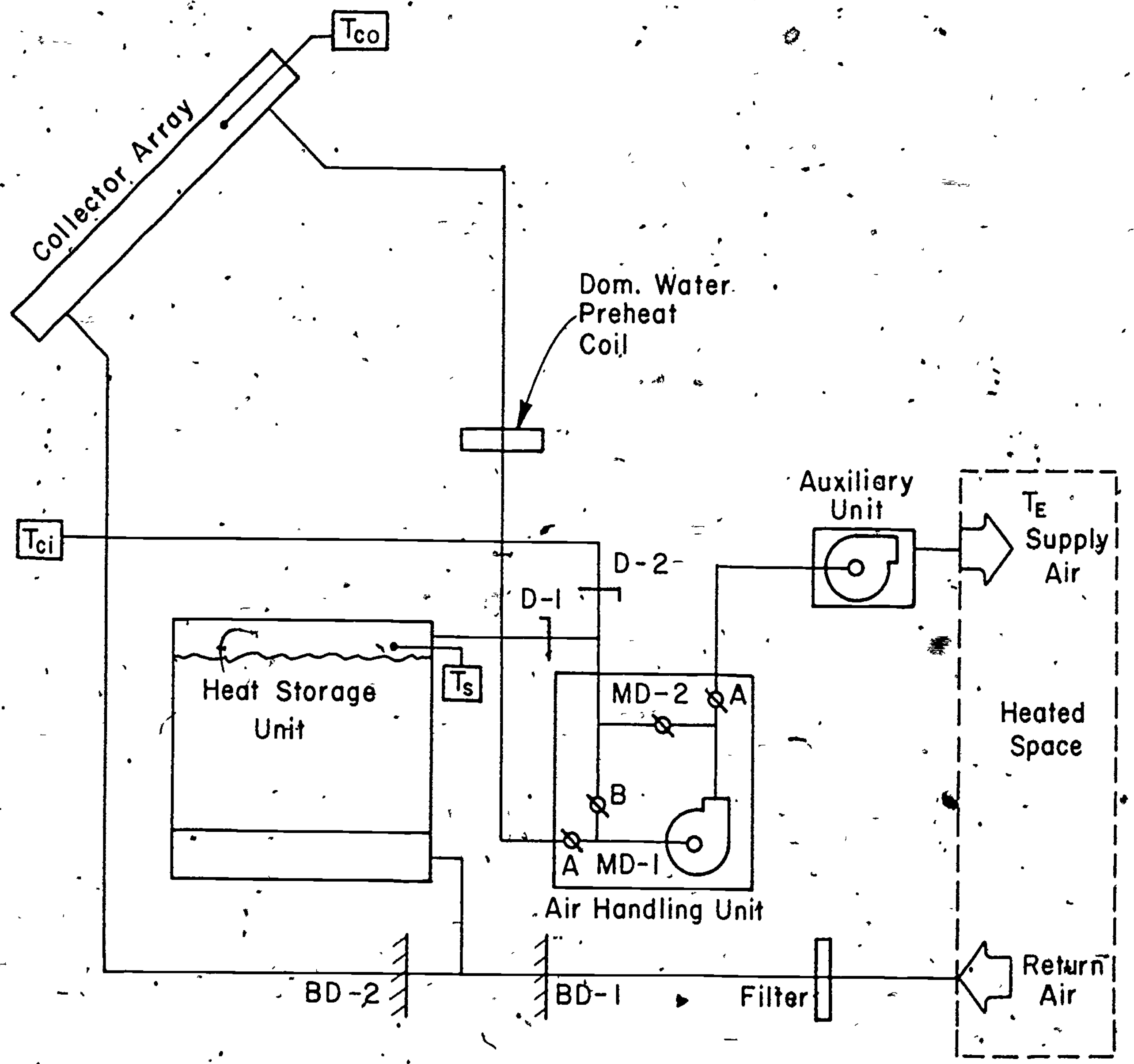

Tco
Collector Array
Dom. Water Preheat Coil
Tci
D-2
D-1
Heat Storage Unit
Ts
MD-2
A
B
A
MD-1
Air Handling Unit
Auxiliary Unit
TE
Supply Air
Heated Space
Return Air
BD-2
BD-1
Filter

Figure 15-1. Schematic Diagram of Solar Air Heating System

MD-2 represent motorized dampers. The function of the controller is to operate the blowers and the motorized dampers to collect and distribute heat for the house. A flowchart of the control strategy is shown in Figure 15-2.

The first decision to be made by the controller concerns whether or not the house needs heat. This is determined by comparing the actual temperature with the desired temperature. If the temperature measured by the temperature sensor in the house indicated that the house does not need heat, then the controller must determine whether or not heat can be stored. This is determined by comparing the collector inlet temperature, T_{ci}, with the collector outlet temperature T_{co}. If T_{co} is not greater than T_{ci}, then obviously there is no heat being collected by the collector and we would not want to circulate the air through the collectors. If T_{co} exceeds T_{ci}, then there is useful energy available and the controller will turn on the blower and the hot water pump and control the dampers to direct the flow through the storage.

If the house requires heat, then the controller must determine whether that heat can be supplied from the collector or from storage. These determinations are again made by comparing temperatures. If T_{co} is not greater than T_{ci}, then obviously we cannot supply the heat from the collector; therefore, the next question to be asked is, can the heat be supplied from storage? This is determined by comparing the storage temperature, T_S, with the reference temperature, T_{SR}, which is nominally set at 100 °F. If T_S is not greater than T_{SR}, then we can not heat from storage; if, however, T_S exceeds T_{SR}, then the controller will turn on the blower and control the dampers to direct the flow through the storage container and to the house by providing heat to the enclosure.

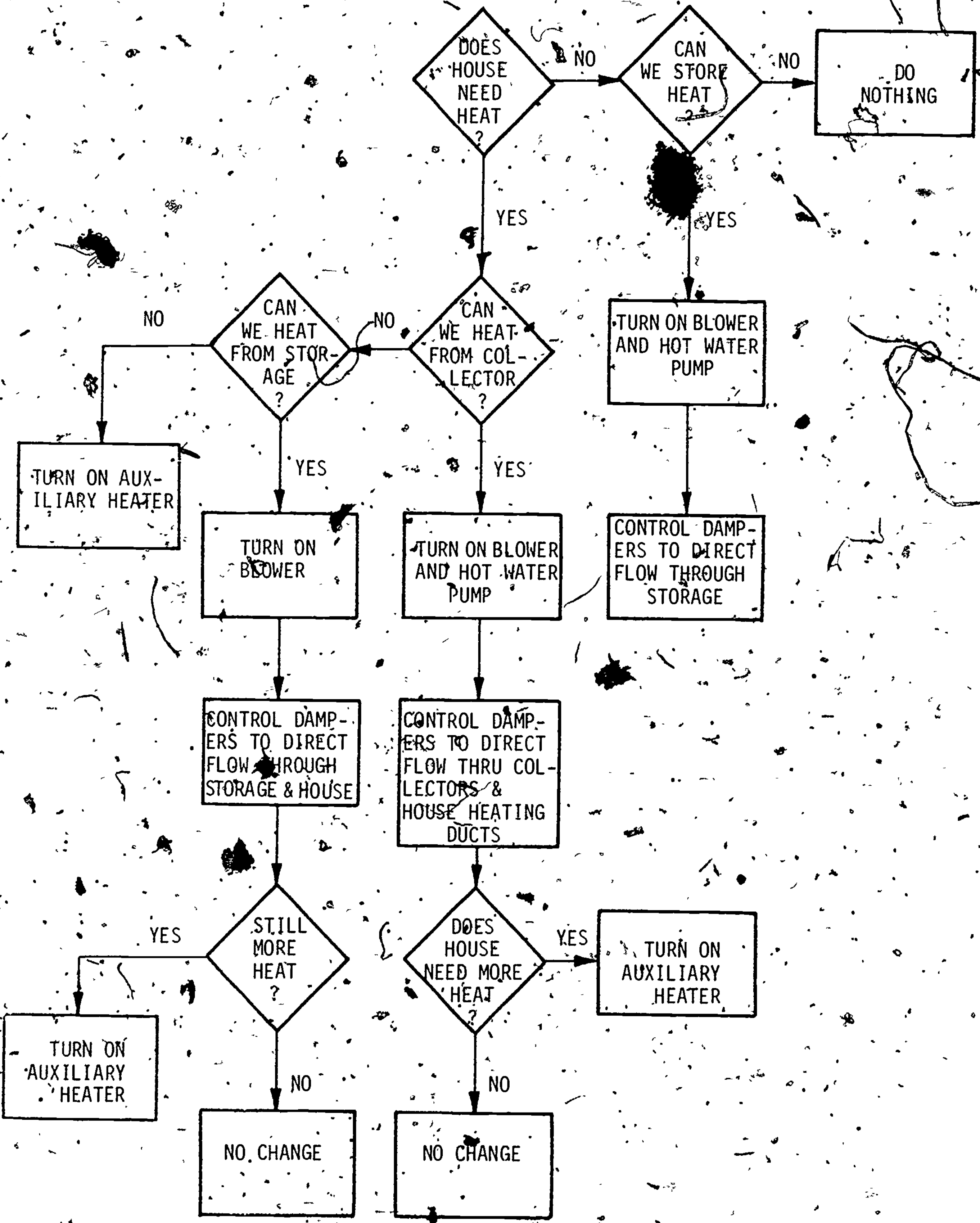

DOES HOUSE NEED HEAT ?
NO
CAN WE STORE HEAT ?
NO
DO NOTHING
YES
YES
TURN ON BLOWER AND HOT WATER PUMP
CONTROL DAMPERS TO DIRECT FLOW THROUGH STORAGE
CAN WE HEAT FROM COLLECTOR ?
NO
CAN WE HEAT FROM STORAGE ?
NO
TURN ON AUXILIARY HEATER
YES
TURN ON BLOWER
YES
TURN ON BLOWER AND HOT WATER PUMP
CONTROL DAMPERS TO DIRECT FLOW THROUGH STORAGE & HOUSE
CONTROL DAMPERS TO DIRECT FLOW THRU COLLECTORS & HOUSE HEATING DUCTS
STILL MORE HEAT ?
YES
TURN ON AUXILIARY HEATER
NO
NO CHANGE
DOES HOUSE NEED MORE HEAT ?
YES
TURN ON AUXILIARY HEATER
NO
NO CHANGE

Figure 15-2. Flow Chart of Control Strategy

It is possible, of course, that the enclosure temperature may continue to decrease due to high heat losses from the house. If the enclosure temperature, T_E, drops below the reference temperature, T_{R2}, then it is clear that still more heat is required. In this case the controller will turn on the auxiliary heater.

If the controller determined that heat could be supplied from the collector, that is, T_{co} is greater than T_{ci}, then the controller will turn on the blower and the hot water pump and control the dampers to direct the flow through the collectors and then to the house heating ducts. It is still possible that this may not supply adequate heating to the house and consequently the controller must compare the enclosure temperature with the second reference temperature to determine whether or not the auxiliary heater must be turned on.

The next step required in the design of a control system is to construct a truth table for the control strategy just described and represented in Figure 15-2. A truth table for this system is shown in Figure 15-3. The left-hand portion of the truth table shows the temperature comparisons. An entry of one in the truth table indicates that the statement is true, whereas a zero represents a situation where it is not true. For example, if T_E is less than T_{R1}, then a one would be entered in the first column in the truth table. The x's represent the "don't care" situation. The middle portion of the truth table shows the various operations. For example, a 1 in the B_{main} column would indicate that the main blower is turned on, whereas a zero would indicate that the blower is turned off. Also, a 1 entry in the MD_I column would indicate that port A on motorized damper number one is open and port B on motorized damper number one is closed. The right-hand portion of the

1	2	3	4	1+3	1	3	1	$1 \cdot 2 + 1 \cdot \bar{3} \cdot \bar{4}$	3	
$T_E<T_{R1}$	$T_E<T_{R2}$	$T_{co}>T_{ci}$	$T_S>T_{SR}$	B_{main}	B_{aux}	MD_1	MD_2	GAS	HWP	MODE
1	0	0	1	1	1	0	1	0	0	Heating from storage
1	1	0	1	1	1	0	1	1	0	Heating from storage plus auxiliary
1	X	0	0	1	1	0	1	1	0	Heating from auxiliary
1	0	1	X	1	1	1	1	0	1	Heating from collector
1	1	1	X	1	1	1	1	1	1	Heating from collector plus auxiliary
0	X	1	X	1	0	1	0	0	1	Store Heat
0	X	0	X	0	0	X	X	0	0	Do Nothing

Figure 15-3. Truth Table for Control Strategy

truth table indicates the mode of operation. For example, consider the first line in the truth table. This line corresponds to the mode where the heat would be supplied from the storage. Suppose that T_E is less than T_{R1} but greater than T_{R2}; then we would enter a one in the first column and a zero in the second column. Suppose furthermore that T_{co} is not greater than T_{ci}; therefore we enter a zero in the third column. Suppose, however, that the storage temperature, T_S, is greater than the reference temperature, T_{SR}; we would want the controller to turn on the blowers and direct the flow through the storage to the house. This flow is directed by controlling motorized dampers one and two. When the blower comes on it will draw air into the return air ducts shown in Figure 15-1, through the filter, through backdraft damper number one, and then into the lower plenum on the storage unit. The air will then flow up through the storage unit, being heated in the process, and exit the storage unit at the top of the plenum. At MD_1 we must have B open and A closed in order that the flow can reach the main blower. The air will exit the main blower and go through MD_2; at MD_2 we must have A open and B closed in order that the flow of air be directed to the supply air ducts. Since it was assumed that T_E was greater than T_{R2}, it was not necessary that the auxiliary unit be turned on and therefore a zero is entered in the gas column in the truth table. Also, since the collectors are not being operated, the hot-water pump will not be turned on and, therefore, there is a zero entered in the HWP column of the truth table. The remaining modes of operation shown on the flow chart of the control strategy are illustrated in the truth table.

The next step required in the design of the control system is to select hardware to implement the logic shown on the truth table. The symbols at the top of the middle portion of the truth table represent the logic that is to be implemented; for example, above the B_{main} column we observe the notation 1 + 3. This indicates that the main blower is to be turned on whenever there is a 1 in column 1 or a 1 in column 3; similarly, the auxiliary blower is to be turned on whenever there is a 1 in column 1, port A on MD_1 is to be opened whenever there is a 1 in column 3, port A on MD_2 is to be opened whenever there is a 1 in column 1, and the gas is to be turned on whenever there is a 1 in columns 1 and 2 or a 1 in column 1 and a zero in columns 3 and 4.

A circuit diagram showing discrete components that may be used to implement the control logic just developed is shown in Figure 15-4. The comparators shown on the left-hand portion of the figure compare the various temperatures throughout the system. These signals are then sent through AND gates, OR gates, and inverters in order to generate the signals that are sent to the motorized dampers, the blowers, and the auxiliary unit.

The solar system designer would not ordinarily concern himself with this level of detail, but would purchase a control unit that accomplishes the above-described tasks; it is important that the designer and the installer understand the functions of the control unit in order to properly conduct system check-out studies and ensure that the solar system operates as desired.

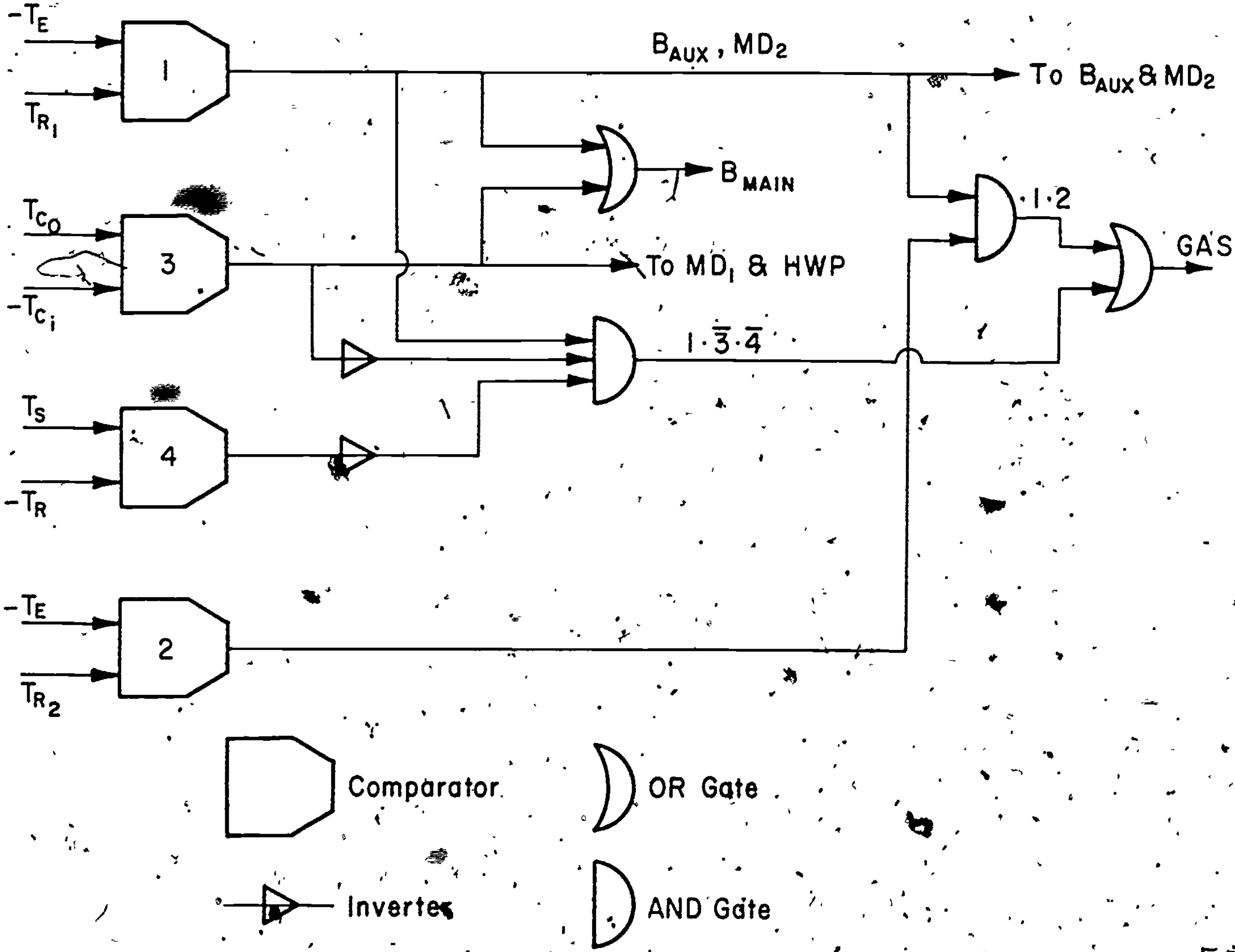
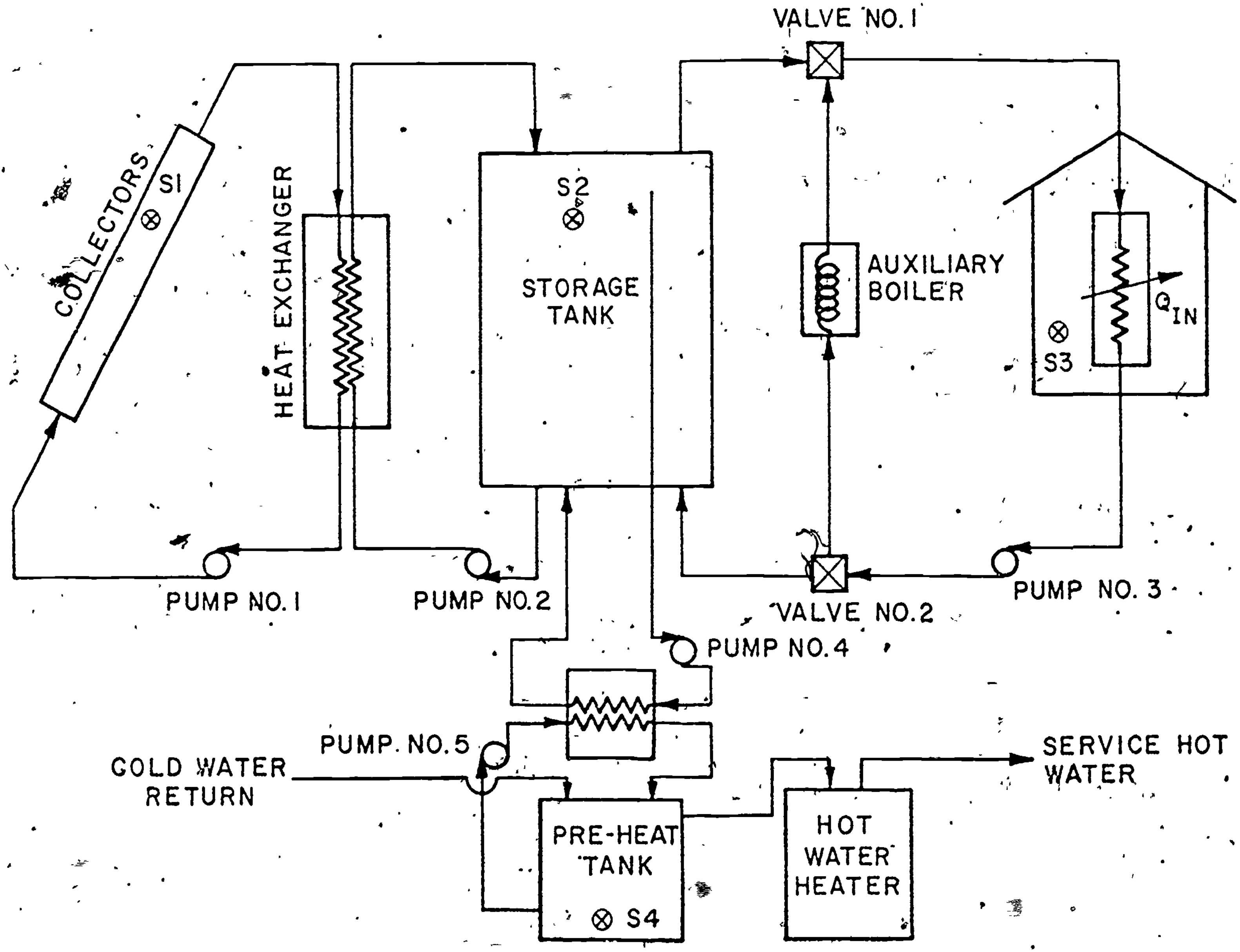

$-T_E$
T_{R_1}
1
B_{AUX}, MD_2
To B_{AUX} & MD_2
B_{MAIN}
T_{C_O}
$-T_{C_i}$
3
To MD_1 & HWP
$1 \cdot 2$
GAS
$1 \cdot \overline{3} \cdot \overline{4}$
T_S
$-T_R$
4
$-T_E$
T_{R_2}
2
Comparator
OR Gate
Inverter
AND Gate

Figure 15-4. Circuit Diagram

HYDRONIC SYSTEM

A sketch of a representative hydronic installation is shown in Figure 15-5. The temperature sensors are indicated by S1, S2, S3 and S4. The signals from these temperature sensors are sent to the control panel in order to control the pumps, labeled P1 through P5, and the valves, labeled V1 and V2. Whenever the temperature recorded at S1 exceeds that at S2 by a preset amount, then the controller starts pumps P1 and P2 and heat is delivered to the storage tank. Also, whenever the temperature at S2 exceeds that at S4 by a preset amount, the controller will start pumps P4 and P5 and heat is supplied to the service hot water preheat tank. Finally, whenever thermostat S3 indicates that the house requires heat, pump P3 is started. The heat is supplied either from the storage tank or from the auxiliary boiler, depending upon the temperature of the water and the storage tank. If the storage tank temperature is not high enough, then the controller will set valves V1 and V2 so that the flow is through the auxiliary boiler and the auxiliary boiler is turned on.

The control for the pumps for the service hot water system operates in a manner similar to that previously described. The sensor S4 senses the temperature of the service hot water in the preheat tank. This is compared with the temperature of water in the storage tank and with a preset maximum value, for example 150 °F. When the temperature in the storage tank exceeds the temperature of the water in the service hot water preheat tank by approximately 10 °F, then pumps P4 and P5 will be started unless the temperature of the water in the preheat tank exceeds 150 °F. We do not want the water in the preheat tank to exceed 150 °F in order to prevent scalding.



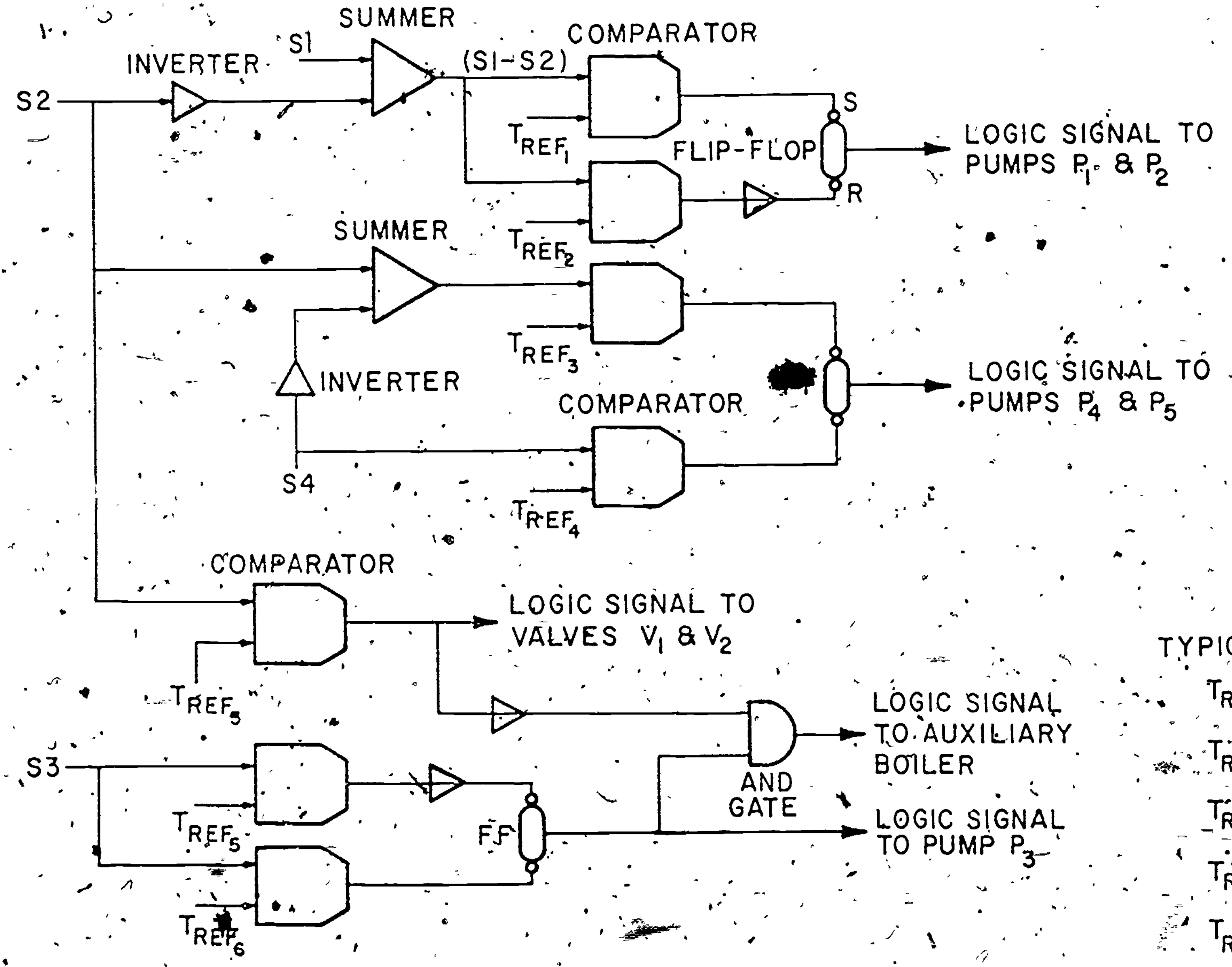
VALVE NO. 1
COLLECTORS
S1
HEAT EXCHANGER
S2
STORAGE TANK
AUXILIARY BOILER
Q_{IN}
S3
PUMP NO. 1
PUMP NO. 2
VALVE NO. 2
PUMP NO. 3
PUMP NO. 4
PUMP NO. 5
COLD WATER RETURN
SERVICE HOT WATER
PRE-HEAT TANK
S4
HOT WATER HEATER

Figure 15-5. Schematic Representation of a Typical Installation.

CONTROL LOGIC, HYDRONIC SYSTEM

A schematic representation of a control logic for a typical installation is shown in Figure 15-6. The system diagrammed in Figure 15-6 operates as follows: The signal from S1 is the temperature being measured at the collectors, while S2 is the temperature in the storage tank. The signals S1 and -S2 (-S2 is obtained by passing the signal S2 through an inverter) are combined in the Summer to produce the signal S1-S2. This is compared with $T_{ref\ 1}$ in the comparator in order to determine whether or not the temperature of the fluid coming out of the collector is high enough to justify turning on the pumps to collect and store the heat. As indicated earlier, this difference should be about 20 °F. When this happens, the logic signal from the comparator is high (+ 5 volts) and this sets the flip-flop, which in turn will turn on pumps P1 and P2. These pumps must be turned off, however, whenever the difference between the collector output temperature and the storage temperature ceases to exceed approximately 4 °F. In order to achieve this, the signal S1-S2 is compared with $T_{ref\ 2}$, then inverted, and then sent to the reset input of the flip-flop. The operation of the flip-flop device is such that when R is high the flip-flop device will send a logic off-signal to pumps P1 and P2. Therefore, when the collector temperature minus the storage temperature is less than 4 °F, the output of the comparator will be low, and consequently, the R input to the flip-flop will be high and this will turn off the pumps.

The control system for supplying heat to the house is shown on the bottom part of Figure 15-6. Thermostat S3 senses the temperature in the house, $T_{ref\ 5}$ represents the lower part of the dead band, and $T_{ref\ 6}$ represents the upper part of the dead band. Whenever the house


SUMMER
S1
INVERTER
S2
(S1-S2)
COMPARATOR
S
T_{REF_1}
FLIP-FLOP
R
LOGIC SIGNAL TO PUMPS P_1 & P_2
SUMMER
T_{REF_2}
T_{REF_3}
INVERTER
LOGIC SIGNAL TO PUMPS P_4 & P_5
COMPARATOR
S4
T_{REF_4}
COMPARATOR
LOGIC SIGNAL TO VALVES V_1 & V_2
T_{REF_5}
LOGIC SIGNAL TO AUXILIARY BOILER
S3
AND GATE
T_{REF_5}
FF
LOGIC SIGNAL TO PUMP P_3
T_{REF_6}

TYPICAL VALUES

T_{REF_1} = 20°F

T_{REF_2} = 4°

T_{REF_3} = 10°

T_{REF_4} = 150°

T_{REF_5} = 65°

T_{REF_6} = 70°

Figure 15-6. Schematic Diagram of the Control Logic for a Typical Installation

temperature drops below $T_{ref\ 5}$, then pump P3 is started. This may be accomplished by comparing S3 with $T_{ref\ 5}$, as shown, then inverting the output of the comparator, and then sending the signal to the set input of the flip-flop. When the temperature in the house reaches the upper part of the dead band, the output of the second comparator will be high, which in turn will cause the flip-flop device to reset itself to turn off pump P3. Finally, the valves V1 and V2 are controlled by the output of the comparator that compares the temperature of the water in the storage tank with $T_{ref\ 3}$. If the storage tank temperature drops below $T_{ref\ 3}$, then valve V2 will direct the flow through the auxiliary boiler. The auxiliary boiler is to be turned on only if pump P3 is on and S2 is smaller than $T_{ref\ 3}$. This is accomplished by passing the two signals through the gate as shown in Figure 15-6.

CONTROL ACTUATORS

The pumps, blowers, valves, and dampers are referred to as the control actuators and produce the desired mechanical operation in response to the electrical control signals. Pumps and blowers are wired through manual switches from the control panel. The switch remains on; it is a safety feature and may even be required in the electrical codes. The switches are to be placed near the motors and not at the control panel.

Control valves and dampers usually require some mechanical adjustment for proper setting. The best practice is to use spring-return two-position dampers or valves which are in the "normal" or most common position when unpowered.

AUXILIARY HEAT CONTROL

The controls on a conventional boiler or forced air furnace must be changed for solar auxiliary purposes. This is because the blower or pump must be actuated from the control panel and not the auxiliary control. Typically, the control panel will produce a second-stage thermostat signal to activate an auxiliary fuel valve of 220 V electrical power relay.

CONTROL SYSTEM CHECK OUT

It is well to check the control system with a "dry" run through the full sequence of modes. The thermostat set points can usually be altered to fake the desired modes. This will assure that the system will "work" when it is first put into operation. There will still need to be adjustments to the control system in order to "tune" it to the highest performance. This amounts to adjustment of set points, dead bands, and anticipators which give the best overall comfort and solar heat utilization.

TRAINING COURSE IN
THE PRACTICAL ASPECTS OF
DESIGN OF SOLAR HEATING AND COOLING SYSTEMS
FOR
RESIDENTIAL BUILDINGS

MODULE 16

SELECTION OF SUBSYSTEM COMPONENTS

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INTRODUCTION

TRAINEE-ORIENTED OBJECTIVE

To be able to select subsystems for a solar heating system.

SUB-OBJECTIVES

At the end of this module the trainee should be able to select:

1. Heat Exchangers
2. Pumps and Blowers
3. Valves, Air Vents, and Dampers

After the collector and storage systems have been sized, it is necessary to select the additional subsystem components. This module considers the selection of heat exchangers, pumps, and valves and air vents for liquid systems, followed by heat exchangers, blowers and pumps, and dampers for air systems. Collectors, storage devices, and controls are discussed in individual modules.

LIQUID SYSTEMS

A schematic representation for a typical liquid system is shown in Figure 16-1. The principal components are the collectors, storage tank, pumps, heat exchangers, valves, and control systems. Additional components that must be included are shown in Figure 16-2. These include the filters, check valve, ion getters, surge tank, and dielectric couplers (if necessary). Methods for selecting these components are presented in the following sections.

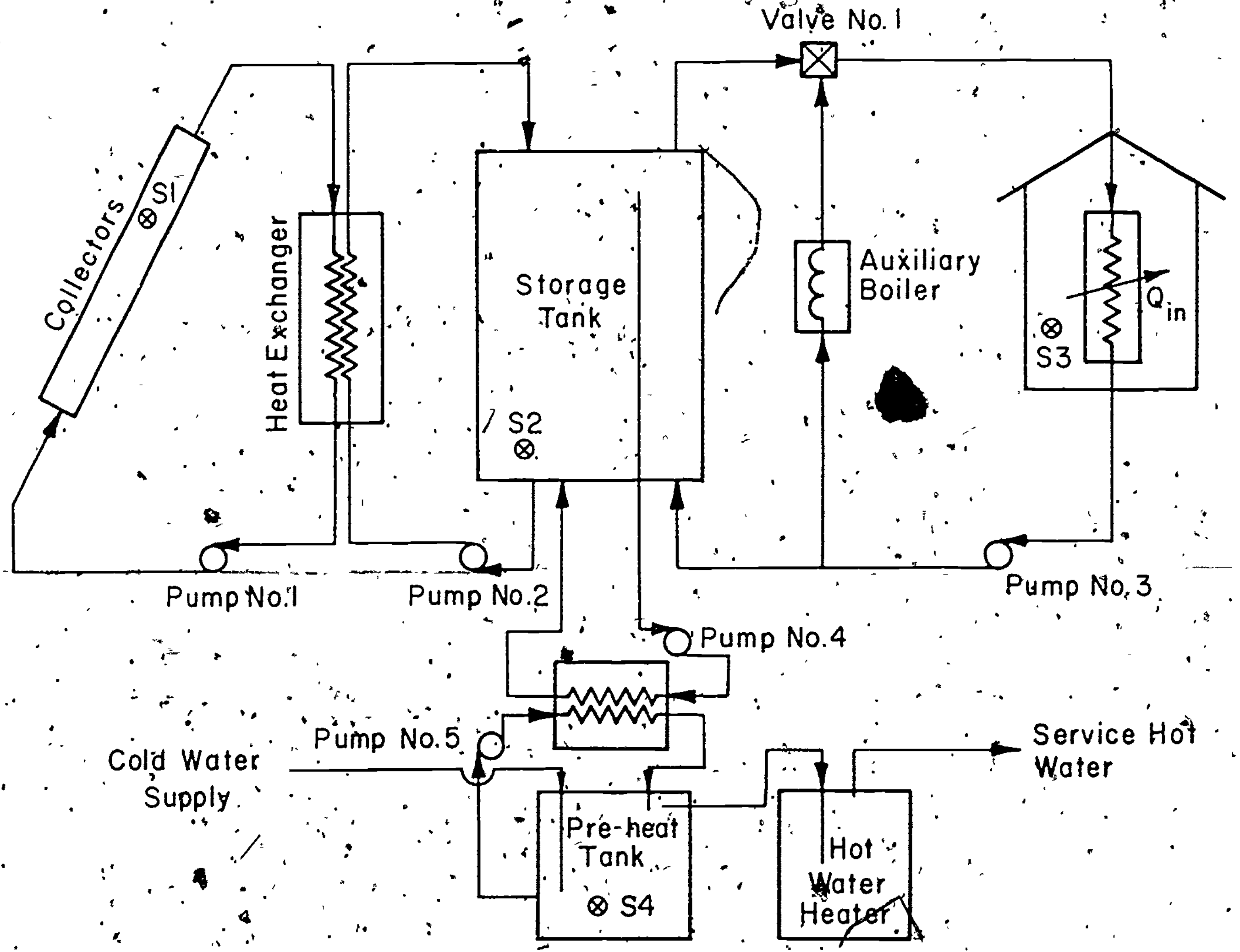

Valve No. 1
Collectors
S1
Heat Exchanger
Storage Tank
S2
Auxiliary Boiler
Q_{in}
S3
Pump No. 1
Pump No. 2
Pump No. 3
Pump No. 4
Pump No. 5
Cold Water Supply
Pre-heat Tank
S4
Hot Water Heater
Service Hot Water

Figure 16-1. Schematic Representation of a Typical Liquid System

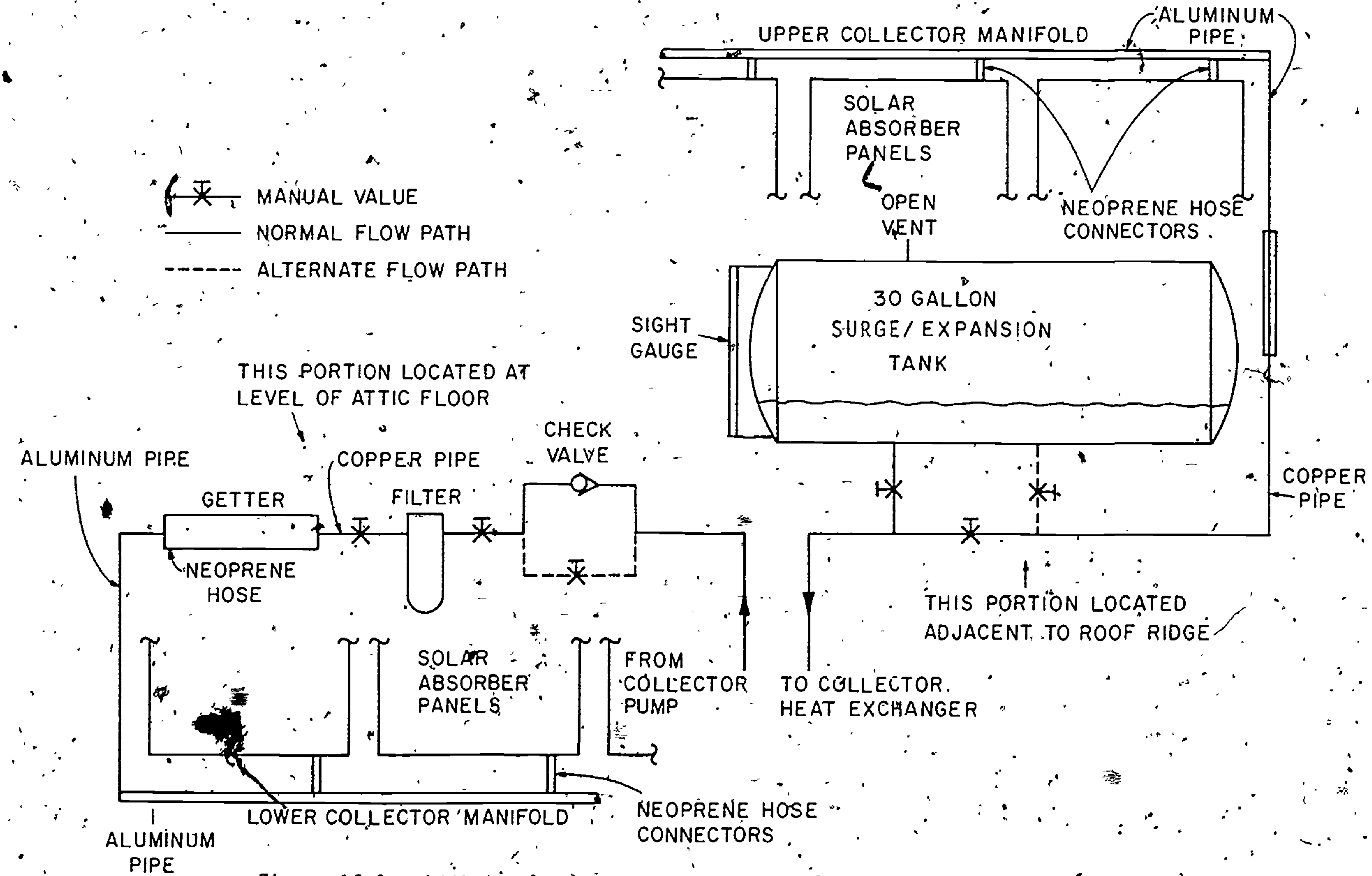

UPPER COLLECTOR MANIFOLD
ALUMINUM PIPE
SOLAR ABSORBER PANELS
OPEN VENT
NEOPRENE HOSE CONNECTORS
MANUAL VALUE
NORMAL FLOW PATH
ALTERNATE FLOW PATH
SIGHT GAUGE
30 GALLON SURGE/EXPANSION TANK
THIS PORTION LOCATED AT LEVEL OF ATTIC FLOOR
CHECK VALVE
ALUMINUM PIPE
COPPER PIPE
GETTER
FILTER
COPPER PIPE
NEOPRENE HOSE
THIS PORTION LOCATED ADJACENT TO ROOF RIDGE
SOLAR ABSORBER PANELS
FROM COLLECTOR PUMP
TO COLLECTOR HEAT EXCHANGER
LOWER COLLECTOR MANIFOLD
NEOPRENE HOSE CONNECTORS
ALUMINUM PIPE

Figure 16-2. Additional Components in a Typical Liquid System

HEAT EXCHANGERS

For liquid-to-liquid heat exchange without mixing of the liquids, shell-and-tube type heat exchangers should be selected. These units are marketed widely and are thus easily obtained. The only uncommon requirement for these heat exchangers is for a low temperature difference between the two liquids. This requirement can be met by using single-pass counterflow design. This is illustrated in Figure 16-3. It can be seen from the temperature profiles in the exchanger that the single-pass counterflow arrangement allows the temperature difference between fluids to be nearly constant along the exchanger. This feature results in a smaller temperature loss across the exchangers. The disadvantage of the single-pass counterflow heat exchanger is its physical dimensions, involving long length and small diameter. A second disadvantage is the high flow rate required through the tubes of the exchanger. The single-pass design results in more tubes in parallel. This means that a higher pumping rate is needed to develop turbulent flow in the tubes. It may be noted that the tube liquid in the collector heat exchanger is the storage tank water and is to be pumped at a high flow rate, as we shall see later. This flow is obtainable with modest pump power because resistance in that loop is low. The high flow rates do, however, nearly eliminate temperature stratification in the storage tank. Highly stratified storage temperatures would thus come at the cost of a larger exchanger or a higher temperature loss.

The information required for selection of a heat exchanger is:

1. Heat load
2. Quantity of fluid entering the heat exchanger (both sides)
3. Specific heat of ~~fluids~~
4. Temperatures in and out (both sides)

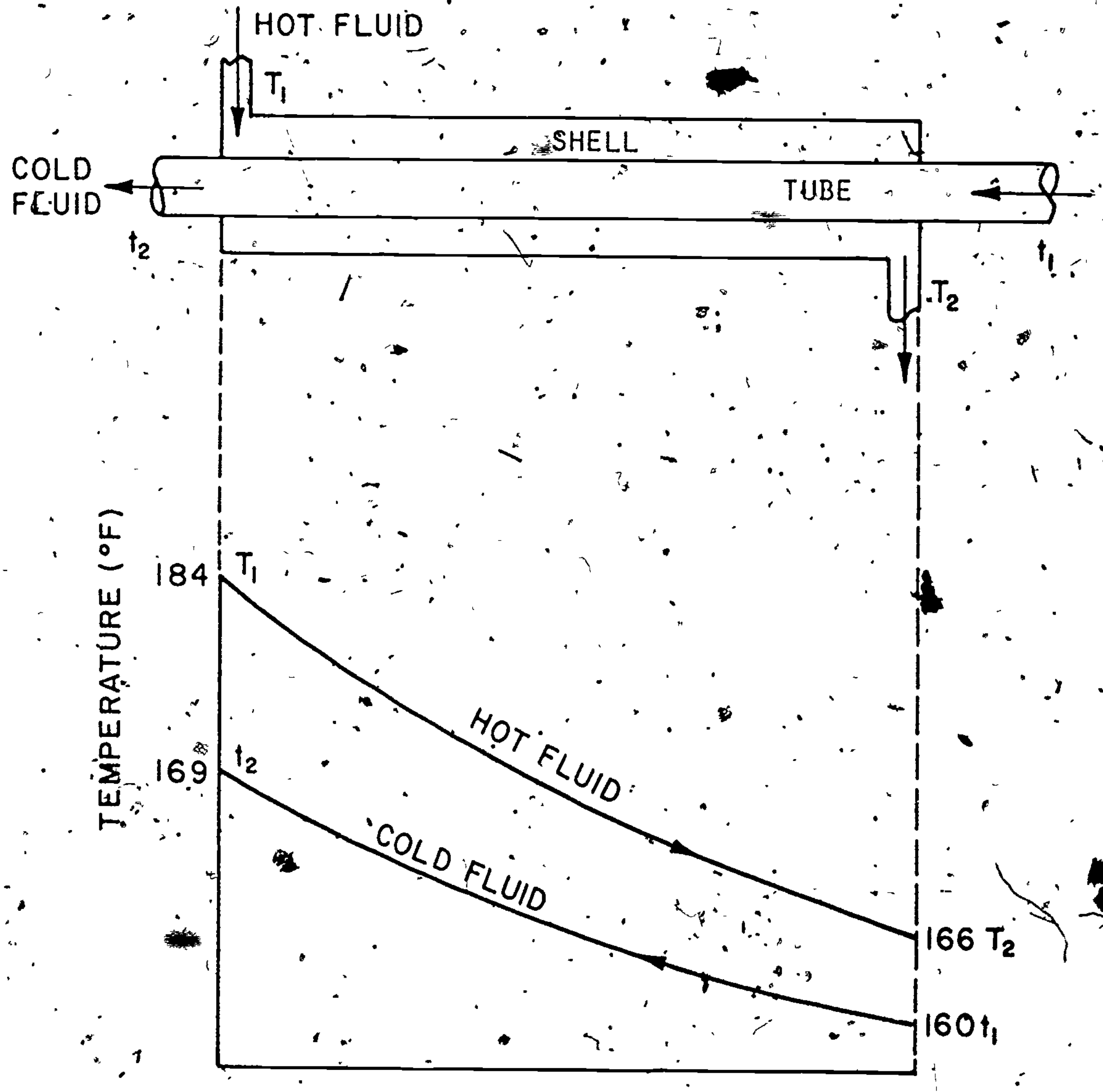

HOT FLUID
T_1
SHELL
COLD FLUID
TUBE
t_2
t_1
T_2
TEMPERATURE (°F)
184 T_1
169 t_2
HOT FLUID
COLD FLUID
166 T_2
160 t_1

Figure 16-3. Single-Pass Counter Flow Heat Exchanger

5. Allowable pressure drops
6. Size limitations (if any)
7. Conditions of corrosion which may affect the system.

Three basic equations are necessary for the solution of the problem of selecting a heat exchanger. The first two equations relate the total heat load to be transferred to the mass flow rate, specific heat, and temperature difference between the entering and leaving fluid for both sides of the heat exchanger. The third equation involves the overall heat transfer coefficient, surface area, and logarithmic mean temperature difference for the heat exchanger. These equations are:

$$Q = (\dot{m}\, C_p)\, \Delta T \quad \text{(shell side)}$$

$$Q = (\dot{m}\, C_p)\, \Delta T \quad \text{(tube side)}$$

$$Q = UA\, (\text{LMTD})$$

where

Q = quantity of heat to be transferred

C_p = specific heat at constant pressure of fluid

$\dot{m}$ = mass flow rate

ΔT = temperature change of fluid

A = surface area

U = overall heat transfer coefficient

LMTD = logarithmic mean temperature difference

The basic heat transfer equation is used to establish the amount of heat exchanger surface required to meet the application requirements. The overall coefficient is a combination of heat transfer rates for both the fluid through the shell side and cooling water through the tubes, and the LMTD is the true, average logarithmic mean temperature difference. The LMTD is defined by

$$\text{LMTD} = \frac{(T_1 - t_2) - (T_2 - t_1)}{\log_e \dfrac{(T_1 - t_2)}{(T_2 - t_1)}}$$

where the temperatures T_1, T_2, t_1, and t_2 are as shown in Figure 16-3 for a single-pass counter flow heat exchanger.

The overall heat transfer coefficient includes the individual fluid film coefficients, the resistance to transmission of heat through the tube walls, and fouling factors for each fluid. These equations may be used to determine the appropriate heat exchanger size for any particular application. The application of these equations to the selection of a particular brand of heat exchangers will be discussed below.

Example: Select a heat exchanger for a solar heating system having 700 ft^2 of collector, an 1100-gallon water-storage tank, and a 60% solution of ethylene glycol and water as the transport medium in the collector loop. The system is to be installed near Fort Collins, Colorado. The mass flow rate through the collector is to be 16 gpm.

Step 1 -- We begin by determining the amount of heat to be transferred. Suppose that we are provided with the following collector characteristics:

$$F_R U_L = 0.86\ Btu/hr\text{-}ft^2\text{-}{}^oF$$

$$F_R \cdot \overline{\tau\alpha} = 0.72$$

$$F_R'/F_R = 0.97$$

Also suppose we design for a condition for which

$$H_T = 365\ Btu/hr\text{-}ft^2$$

and

$$T_i - T_a = 100\ {}^oF.$$

Then the amount of heat collected per square foot of collector is:

$$Qu/A = (350)\ (.698) - (.8342)\ (100)$$

$$= 172 \text{ Btu/hr-ft}^2$$

Therefore, the total heat collected is

$$Q = 113{,}000 \text{ Btu/hr.}$$

Step 2 -- We next determine the temperature changes through the shell side and the tube side of the heat exchanger. The highest temperature fluid should be circulated through the shell side. The specific heat of the mixture is about 0.8 $\frac{\text{Btu}}{\text{lb}}$. (It varies with temperature and this is an average value.)

Therefore,

$$(\dot{m}\ Cp)_S = \left(16\ \frac{\text{gal}}{\text{min}}\right)\left(8.25\ \frac{\text{lb}}{\text{gal}}\right)\left(0.8\ \frac{\text{Btu}}{\text{lb-}^{\text{o}}\text{F}}\right)\left(60\ \frac{\text{min}}{\text{hr}}\right)$$

$$= 6336\ \frac{\text{Btu}}{\text{hr-}^{\text{o}}\text{F}}$$

Therefore the temperature drop on the shell side is

$$\Delta T_S = 113000/6336 = 18\ ^{\text{o}}\text{F}.$$

Now suppose that the mass flow rate through the tube side is 25 gal/min. Then:

$$(\dot{m}\ Cp)_T = \left(25\ \frac{\text{gal}}{\text{min}}\right)\left(8.33\ \frac{\text{lb}}{\text{gal}}\right)\left(1\ \frac{\text{Btu}}{\text{lb-}^{\text{o}}\text{F}}\right)\left(60\ \frac{\text{min}}{\text{hr}}\right) = 12500\ \frac{\text{Btu}}{\text{hr-}^{\text{o}}\text{F}}$$

Therefore,

$$\Delta T_T = 113000/12500 = 9\ ^{\text{o}}\text{F}$$

Step 3 -- Next we evaluate the LMTD. This is made simple by referring to Table 16-1. We calculate the small temperature difference (STD) and the large temperature difference (LTD) from

$$STD = \min \left[(T_1 - t_2),\ (T_2 - t_1) \right]$$

$$LTD = \max \left[(T_1 - t_2),\ (T_2 - t_1) \right]$$

where

T_1 = temperature of fluid entering shell side

T_2 = temperature of fluid leaving shell side

t_1 = temperature of fluid entering tube side

t_2 = temperature of fluid leaving tube side.

In this example we shall suppose that

$$T_1 = 184\ ^oF,$$

$$t_1 = 160\ ^oF$$

Then

$$T_2 = 184-18 = 166\ ^oF$$

and

$$t_2 = 160+9 = 169\ ^oF.$$

Also,

$$T_1 - t_2 = 15\ ^oF$$

and

$$T_2 - t_1 = 6\ ^oF.$$

Therefore,

$$LTD = 15,\ STD = 6$$

and

$$STD/LTD = 0.4.$$

From Table 16-1, LMTD/LTD = 0.655. Hence, LMTD = 9.8 ^oF.

Step 4 -- Using the heat transfer equation, Q = UA (LMTD) and a heat transfer coefficient (nominal) of 275 Btu/hr-ft^2-^{o}F, we obtain

$$A = \frac{113000}{(275)(9.8)} = 42 \text{ ft}^2.$$

Step 5 -- Use Table 16-2 to select a heat exchanger having approximately the area determined in Step 4. From Table 16-2, we can select a model SSF-603 with Y tubes or an F-603 with Y tubes.

Step 6 -- We now select the baffle spacing. We do this by determining shell and tube velocity factors by referring to Table 16-3 and then computing velocities according to:

shell velocity (ft/sec) = (shell velocity factor) x (gpm through shell side)

tube velocity (ft/sec) = (tube velocity factor) x (gpm through tube side) x (number of passes).

For best results the velocities should be between 1 and 6 feet per second.

From Table 16-3 we obtain

SVF	Velocity (ft/sec)	TVF	Velocity (ft/sec)
.244	3.9	.038	.825
.122	1.95		
.061	0.98		
.031	0.50		

We can select either an H or D baffle and obtain an appropriate velocity through the shell side. The tube velocity is slightly low and could be improved by increasing the flow rate.

PUMPS

The pumps must be selected to provide the required design flow rates. The pumps should be centrifugal type with direct coupled motors. Centrifugal pumps provide two advantages over the positive displacement style pump. Centrifugal pumps offer a safety feature in that they will pump only a small amount above rated pressure if the fluid loop should be blocked. Thus, such an occurrence would neither damage the pump nor burst a fluid line. A second advantage, particularly on the collector loop, is the increase in flow rate as the temperature of the fluid increases. This is due to viscosity changes of the fluid and improves the collector efficiency at high temperatures.

Pumps are rated according to the flow rate they will provide when subjected to a given head pressure. The head pressure is calculated by determining the pressure drop in each component of a loop. For example, consider the loop for pump P1 in Figure 16-1. The pressure drop consists of a Δp in the collectors, a Δp due to pipe friction, a Δp due to elbows, a Δp due to valves, a Δp due to the heat exchanger, a Δp due to the filter, and a Δp due to the ion getter. In a closed loop that is kept full, one does not include the head pressure due to pumping the fluid to a higher elevation. However, in a system that is drained, this head would have to be included.

The pressure drop through the collectors will represent the major drop in the system. This will have to be provided by the collector manufacturer or otherwise determined experimentally before the system design can be completed. Typical pressure drops are on the order of 2 $\mathrm{lb/in^2}$ per collector panel. This can be related to feet of water by the relation:

1 ft H_2O = 0.433 psi.

The pressure drop through copper pipes can be determined by referring to Figure 16-4, which gives pressure drop in various sizes of pipe at various flow rates. For example, at a flow rate of 15 gpm through 1 1/4 inch copper pipe, the Δp is 5 feet of water per 100 feet of pipe. Similar charts may be obtained for other types of pipe.

The Δp through elbows, valves, and tees may be determined by referring to Figure 16-5, or by simply adding about 50% to the Δp for the pipes.

The pressure drop through the ion getter is so small that it can be ignored. However, the Δp through the filter can be significant and should be obtained from the manufacturer. In the case of the filter in CSU Solar I the values range between those shown in Table 16-4.

The Δp through the heat exchanger can be obtained from the manufacturer or calculated by the following procedure (from Young Radiator Co.).

The set of curves shown in Figures 16-6 through 16-9 can be used to obtain pressure drops on the shell and tube sides of a heat exchanger. The curves are:

PD1 - The pressure drop through the shell side of the tube bundle utilizing water.

PD2 - The pressure drop through the tube side of the tube bundle and the drop due to the change of direction in multi-pass Heat Exchangers utilizing water.

PD3 - The pressure drop resulting from sudden contraction and expansion as occurs between the entrance and exit connections and the tube bundle on the shell side; and between the channel or bonnet, and the tube bundle and entrance and exit connections on the tube side.

PD4 - The velocity through a round opening or pipe of a given diameter at the given flow rate.

To obtain the shell side pressure drop through a heat exchanger:

1. Calculate the shell velocity and enter either PD1 for water or PD2 for oil and obtain P_w(water) or P_o(oil).
2. Multiply, as indicated on graphs PD1 or PD2, P_w or P_o by the tube bundle length in inches, by the ratio of the inside shell diameter divided by the baffle spacing, and by the indicated correction factor.
3. Record the result as the pressure drop through the tube bundle in psi.
4. Obtain the entrance velocity from PD5 by entering with the flow in gpm and the size of the connection opening.
5. Calculate the ratio of the entrance velocity from Step 4 to the shell velocity from Step 1. Divide the larger of the two by the smaller so that the ratio is larger than one.
6. Enter PD4 with the ratio from Step 5 and multiply the factor obtained by the square of the smaller velocity from Step 1 or Step 4, whichever is appropriate. Multiply this result by .88 for oil.
7. Record this result as the pressure drop through the entrance and exit connection in psi.
8. Add the figures from Step 3 and Step 7 to obtain the total pressure drop in psi through the shell side of the heat exchanger.

9. Obtain the pressure drop factor for entering tube loss by entering PD4 with the ratio of the tube velocity from Step 1 to the channel velocity from Step 7. Divide the larger velocity by the smaller so that the ratio is larger than one.

10. Obtain the pressure drop factor for the entrance loss by entering PD4 with the ratio of the entrance velocity from STEP 8 to the channel velocity from Step 7. Divide the larger velocity by the smaller so that the ratio is larger than one.

11. Add the values from Step 9 and Step 10 to .0625.

12. Multiply the value from Step 11 by the square of the channel velocity from Step 7.

13. Record this result as the pressure drop through the entering and exit connections.

14. Add the values from Step 6 and Step 13 to obtain the total pressure drop in psi through the tube side of a heat exchanger.

After determining the individual pressure drops, the total Δp may be determined in order to select a pump. For example, in CSU Solar I, the total pressure drop in the collector loop is found to be ~47 feet of water. This was made up of the following items.

$$\Delta p_{pipes} = (140\ ft)\left(\frac{6\ ft\ H_2O}{100\ ft}\right) = 8.4\ ft$$

$$\Delta p_{elbows} = \left(8\ \frac{ft}{elbow}\right)(13\ elbows)\left(\frac{6\ ft\ H_2O}{100\ ft}\right) = 6.24\ ft$$

$$\Delta p_{valves} = \left(1\ \frac{ft}{valve}\right)(12\ valves)\left(\frac{6\ ft\ H_2O}{100\ ft}\right) = 0.72\ ft$$

$$\Delta p_{ht.\ exc.} = 20.8\ ft$$

$\Delta p_{coll} = 9.2$ ft

$\Delta p_{filter} \approx 2.3$ ft.

A typical pump selection chart is shown in Figure 16-10. The pump should be selected to provide the desired flow rate with the Δp calculated.

VALVES AND AIR VENTS

It is very important that valves and air vents be included in a liquid system. The purpose of the valves is to provide for the proper flow rate and uniform flow through the collectors. The valves should be adjusted after the system is installed to insure that the flow rates are close to those required by the design. Also, since it is virtually impossible to keep air out of an unpressurized system, it is absolutely essential that air vents be included in the system design. The air vents and valves should be made of the same material as the plumbing that they connect to in order not to add to the corrosion problem.

In addition to the valves required for flow regulation, many systems (such as that illustrated in Figure 16-1) will require some three-way valves. These valves are a part of the control system and are used to direct the flow through the auxiliary boiler.

AIR SYSTEMS

We will not spend nearly as much time discussing air systems as we did water systems because the principal manufacturers of air collectors manufacture and sell a complete system. Also, the major problems are encountered more with respect to installation than with respect to design.

HEAT EXCHANGERS

The only heat exchanger used in a typical air system is an air-to-water heat exchanger used to provide for service hot water. This can consist of a copper coil inside the air duct on the return side of the collector. This can be sized quite easily by considering the amount of service hot water to be provided, the air flow rate and temperature through the air duct, the water flow rate through the coil, and the heat transfer coefficient between the air and the coil. The equations required to solve this heat transfer problem are:

$$\dot{Q} = (\dot{m}c_p)_{air}\ \Delta T_{air}$$

$$\dot{Q} = (\dot{m}c_p)_w\ \Delta T_w$$

$$Q = kA\ (\bar{T}_{air} - \bar{T}_w)$$

where $\bar{T}_a$ and $\bar{T}_w$ represent the average air temperature and water temperature across and through the coil, and k represents the heat transfer coefficient, and A represents the area of the coil.

The problem now is that we have more unknowns than we have equations. The difficulty is that the temperature of the water into the coil is unknown and is a function of the dynamics of the preheat tank. It may be determined from a differential equation for the heat balance of the preheat tank. Time does not permit us to develop general parameterized solutions to this dynamic system's problem. However, since the cost of this heat exchanger is extremely minor relative to the cost of other components in this system, and the performance of the heat exchanger and service hot water preheat tank system is not terribly sensitive to the heat exchanger area, it is recommended that a standard size be used in residential applications. For example, Solaron Corporation supplies a standard size of 15" x 18" which is used in systems having flow rates up to 1300 CFM.

BLOWERS AND PUMPS

The only pumps required in a typical air system are the pumps required for the service hot water. Since these pumps interact with the service hot water, they should be bronze pumps rather than have iron castings. A quite satisfactory pump has been found to be the March 809 BF.

The blower should be sized in a manner similar to that which was used for sizing pumps in liquid systems. That is, the static pressure lost throughout the system must be calculated based on the desired flow rates, and this is used to size the blower. The pressure drop may be determined from Figure 16-11 which gives friction loss in straight ducts for flow rates in the range of interest for residential applications. The major pressure drop in the system will be that through the collectors and this would have to be provided by the collector manufacturer. As with the liquid system, the ΔT's through each component in a loop must be determined, and the total pressure drop consists of the sum of the individual pressure drops. This is then used to select the blowers.

It is important in an air system that the flow rate required by the auxiliary furnace be matched to that required for the solar part of the system. If this is not the case -- for example, if the solar part requires a flow rate of 600 CFM and the auxiliary furnace requires a flow rate of 1000 CFM -- then additional ducting and controls must be provided. This obviously increases the cost of the system.

Table 16-1. Logarithmic Mean Temperature Difference Factors*

STD ÷ LTD	LMTD ÷ LTD	STD ÷ LTD	LMTD ÷ LTD	STD ÷ LTD	LMTD ÷ LTD	STD ÷ LTD	LMTD ÷ LTD
1	2	1	2	[illegible]	2	1	2
		0.25	0.541	0.50	0.721	0.75	0.870
0.01	0.215	0.26	0.549	0.51	0.728	0.76	0.874
0.02	0.251	0.27	0.558	0.52	0.734	0.77	0.879
0.03	0.277	0.28	0.566	0.53	0.740	0.78	0.886
0.04	0.298	0.29	0.574	0.54	0.746	0.79	0.890
0.05	0.317	0.30	0.582	0.55	0.753	0.80	0.896
0.06	0.334	0.31	0.589	0.56	0.759	0.81	0.902
0.07	0.350	0.32	0.597	0.57	0.765	0.82	0.907
0.08	0.364	0.33	0.604	0.58	0.771	0.83	0.913
0.09	0.378	0.34	0.612	0.59	0.777	0.84	0.918
0.10	0.391	0.35	0.619	0.60	0.783	0.85	0.923
0.11	0.403	0.36	0.626	0.61	0.789	0.86	0.928
0.12	0.415	0.37	0.634	0.62	0.795	0.87	0.934
0.13	0.427	0.38	0.641	0.63	0.801	0.88	0.939
0.14	0.438	0.39	0.648	0.64	0.806	0.89	0.944
0.15	0.448	0.40	0.655	0.65	0.813	0.90	0.949
0.16	0.458	0.41	0.662	0.66	0.818	0.91	0.955
0.17	0.469	0.42	0.669	0.67	0.823	0.92	0.959
0.18	0.478	0.43	0.675	0.68	0.829	0.93	0.964
0.19	0.488	0.44	0.682	0.69	0.836	0.94	0.970
0.20	0.497	0.45	0.689	0.70	0.840	0.95	0.975
0.21	0.506	0.46	0.695	0.71	0.848	0.96	0.979
0.22	0.515	0.47	0.702	0.72	0.852	0.97	0.986
0.23	0.524	0.48	0.709	0.73	0.858	0.98	0.991
0.24	0.533	0.49	0.715	0.74	0.864	0.99	0.995

* Young: Fixed Tube Bundle Heat Exchangers, Catalog No. 1275, (Wisconsin: Young Radiator Co., 1975), p. 7.

Table 16-2. Surface Area*

TUBE SURFACE (square feet)

SERIES	Type HF-SSF	Type F-HP		Type SSF			
	One Pass	One, Two & Four Pass		One & Two Pass		Four Pass	
	Y TUBES	Y TUBES	R TUBES	Y TUBES	R TUBES	Y TUBES	R TUBES
201	1.5	---	---	---	---	---	---
202	3.0	---	---	---	---	---	---
301	---	3.6	2.6	3.9	2.8	3.6	---
302	---	7.4	5.2	7.9	5.8	7.2	---
303	---	11.1	7.9	11.9	8.7	10.9	---
502	---	18.1	11.2	18.4	11.8	16.2	11.5
503	---	27.1	16.8	27.8	17.8	25.0	17.4
504	---	36.1	22.4	37.3	23.9	33.4	23.3
602	---	26.9	17.5	27.3	18.3	25.6	16.6
603	---	40.2	26.2	41.1	27.6	38.5	25.0
604	---	53.6	34.9	55.0	36.9	51.5	33.4
606	---	80.8	52.7	83.2	55.8	77.9	50.6
608	---	107.7	70.2	111.1	74.5	104.1	67.5
802	---	---	34.1	---	32.7	---	32.7
803	---	---	50.6	---	49.2	---	49.2
804	---	---	67.1	---	65.7	---	65.7
805	---	---	83.6	---	82.2	---	82.2
806	---	---	100.1	---	98.6	---	98.6
807	---	---	116.6	---	115.1	---	115.1
808	---	---	133.1	---	131.6	---	131.6
809	---	---	149.6	---	148.1	---	148.1
810	---	---	166.1	---	164.6	---	164.6

---

* Young: Fixed Tube Bundle Heat Exchangers, Catalog No. 1275, (Wisconsin: Young Radiator Co., 1975), p. 8.

Table 16-3. Velocity Factors*

SHELL VELOCITY FACTORS

TUBE SIZE OD		Y-1/4 INCH				R-3/8 INCH			
Baffle Spacing (in.)		H 1-1/8	D 2-1/4	E 4-1/2	A 9	H 1-1/8	D 2-1/4	E 4-1/2	A 9
	200	.74	---	---	---	---	---	---	---
	300	.420	.210	.105	---	.472	.236	.118	---
SERIES	500	.297	.148	.074	0.37	.333	.166	.083	.042
	600	.244	.122	.061	.031	.279	.140	0.70	.035
	800	---	---	---	---	---	.104	.052	.026

TUBE VELOCITY FACTORS

TUBE SIZE OD		Y-1/4 INCH				R-3/8 INCH			
Baffle Spacing (in.)		H 1-1/8	D 2-1/4	E 4-1/2	A 9	H 1-1/8	D 2-1/4	E 4-1/2	A 9
	200	.29	---	---	---	---	---	---	---
SERIES	300	.117	.117	.117	---	.107	.107	.107	---
ONE	500	.049	.049	.049	.049	.051	.051	.051	.051
PASS	600	.033	.033	.033	.033	.032	.032	.032	.032
	800	---	---	---	---	---	.017	.017	.017

* Young: Fixed Tube Bundle Heat Exchangers, Catalog No. 1275, (Wisconsin: Young Radiator Co., 1975), p. 8.

Table 16-4. Pressure Drops through Filter in CSU Solar I

	ΔP (psi)		
	Fine	Medium	Coarse
16 gpm 205 °F 60% ethylene glycol (in water)	2.10	0.70	0.35
5 gpm 87 °F pure water	0.60	0.20	0.10

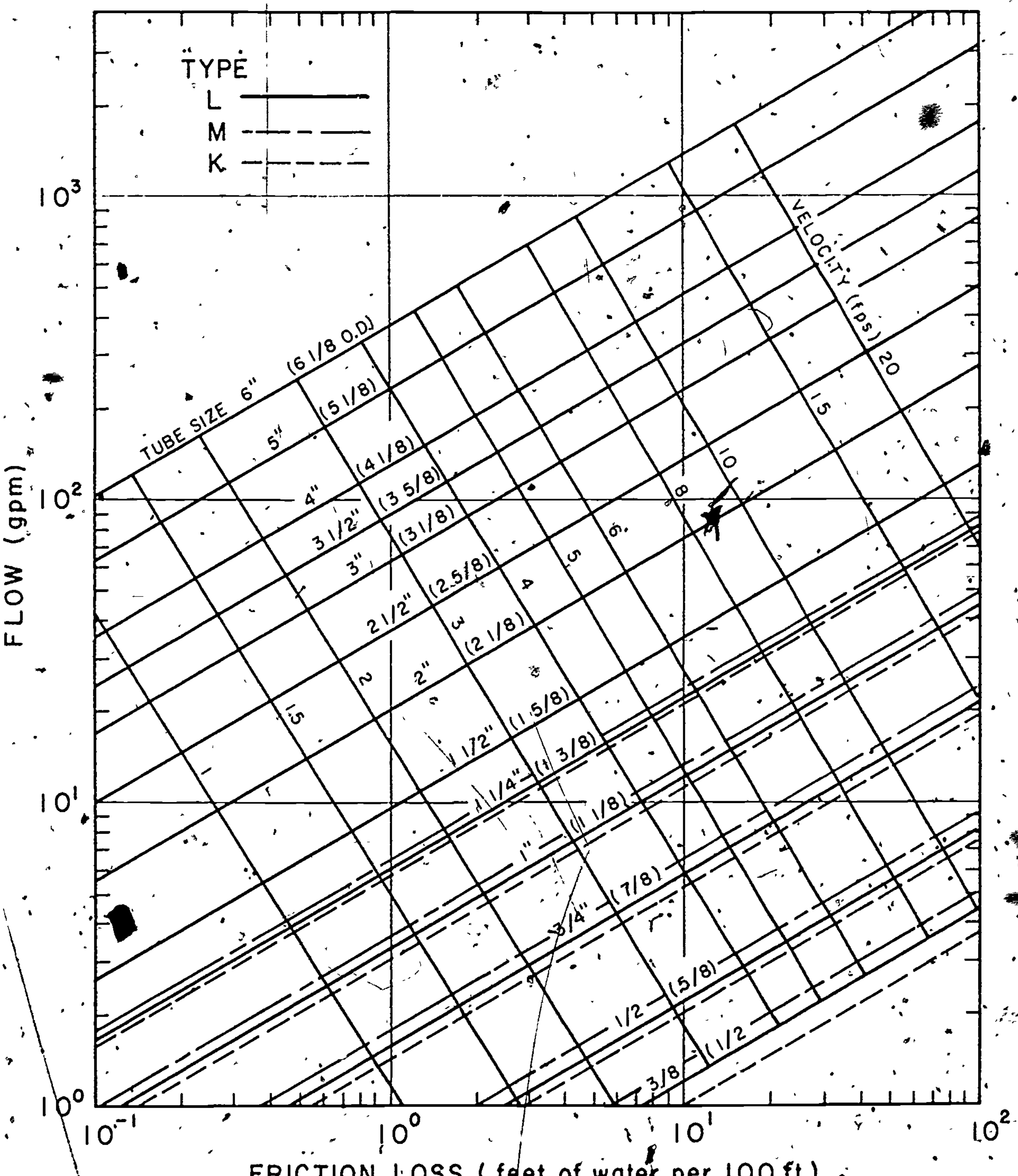
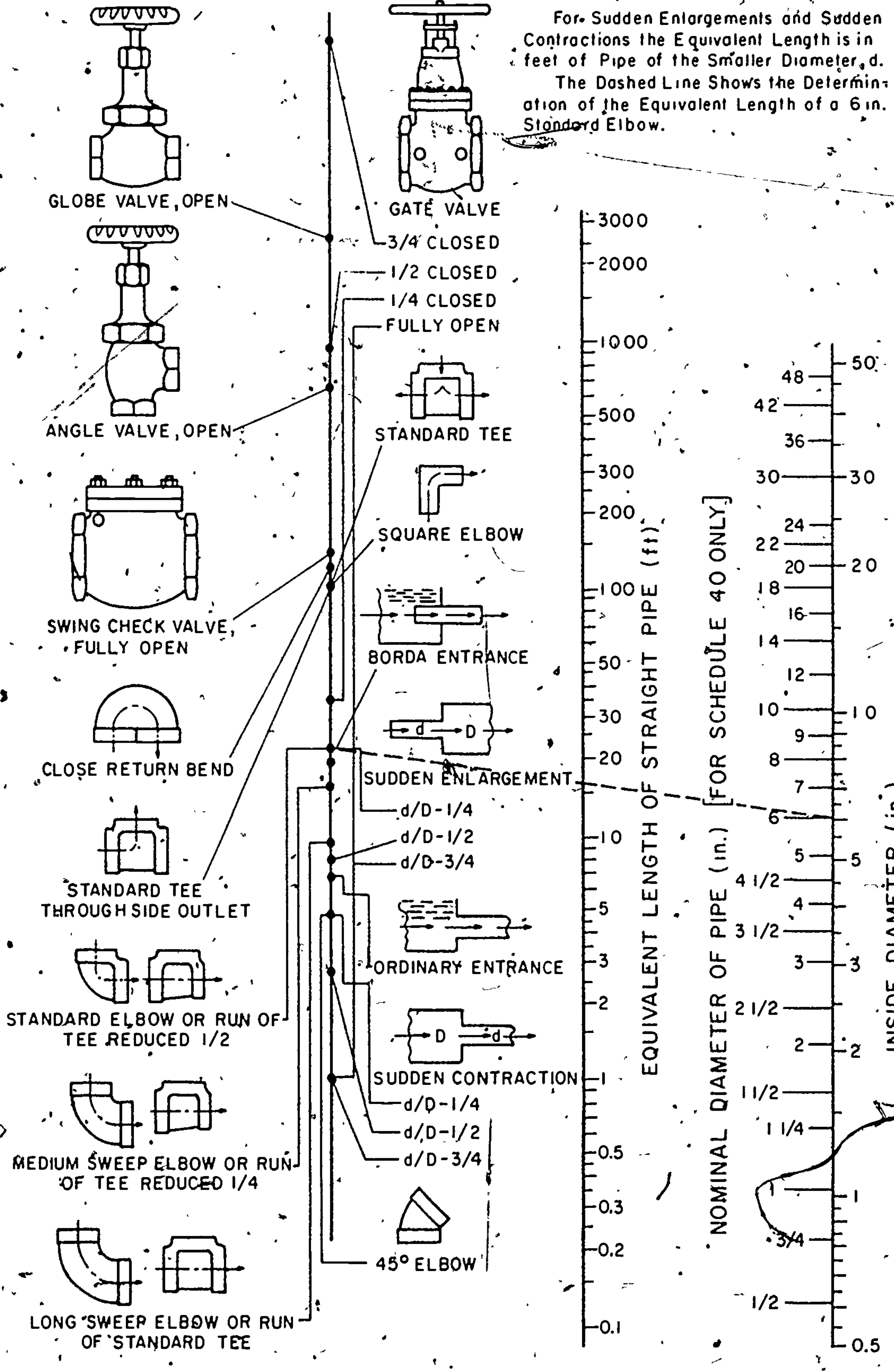


Figure 16-4. Friction Loss in Copper Tubing



For Sudden Enlargements and Sudden Contractions the Equivalent Length is in feet of Pipe of the Smaller Diameter, d.
The Dashed Line Shows the Determination of the Equivalent Length of a 6 in. Standard Elbow.

Figure 16-5. Pressure Loss in Various Elements.

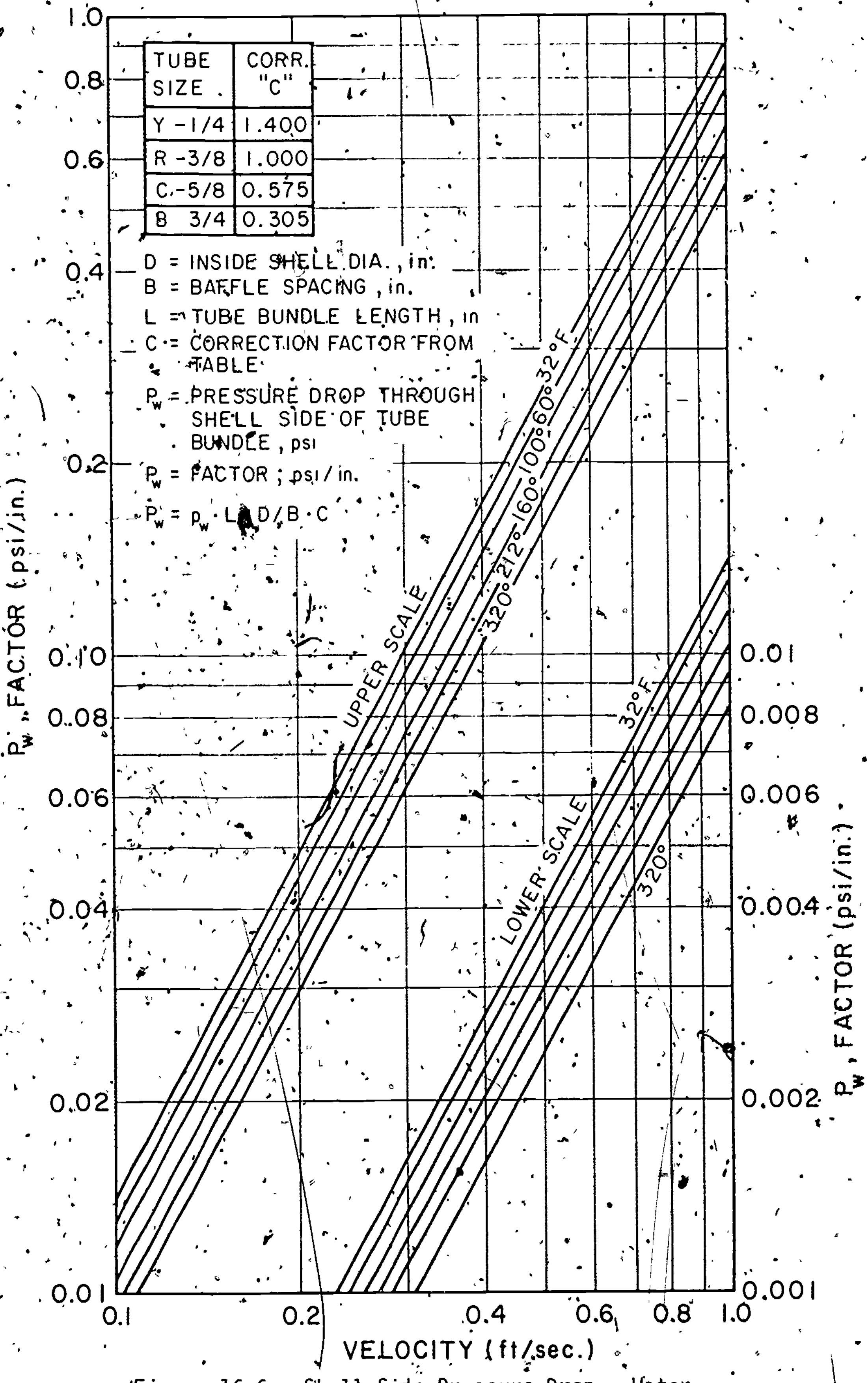
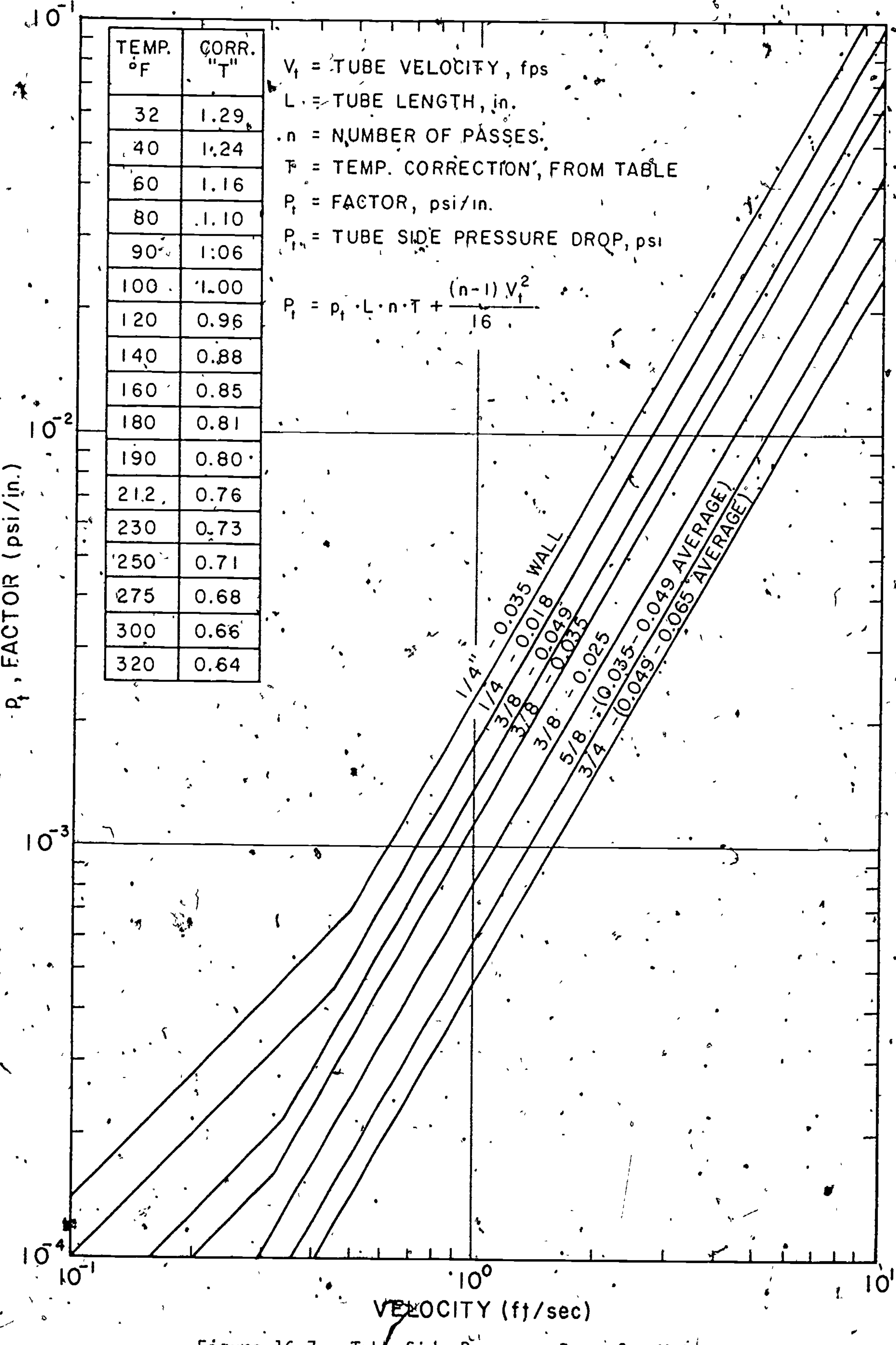


Figure 16-6. Shell Side Pressure Drop - Water.



TEMP. °F	CORR. "T"
32	1.29
40	1.24
60	1.16
80	1.10
90	1.06
100	1.00
120	0.96
140	0.88
160	0.85
180	0.81
190	0.80
212	0.76
230	0.73
250	0.71
275	0.68
300	0.66
320	0.64

V_t = TUBE VELOCITY, fps
L = TUBE LENGTH, in.
n = NUMBER OF PASSES
T = TEMP. CORRECTION, FROM TABLE
p_t = FACTOR, psi/in.
P_t = TUBE SIDE PRESSURE DROP, psi

$$P_t = p_t \cdot L \cdot n \cdot T + \frac{(n-1)\, V_t^2}{16}$$

Figure 16-7. Tube Side Pressure Drop for Water.

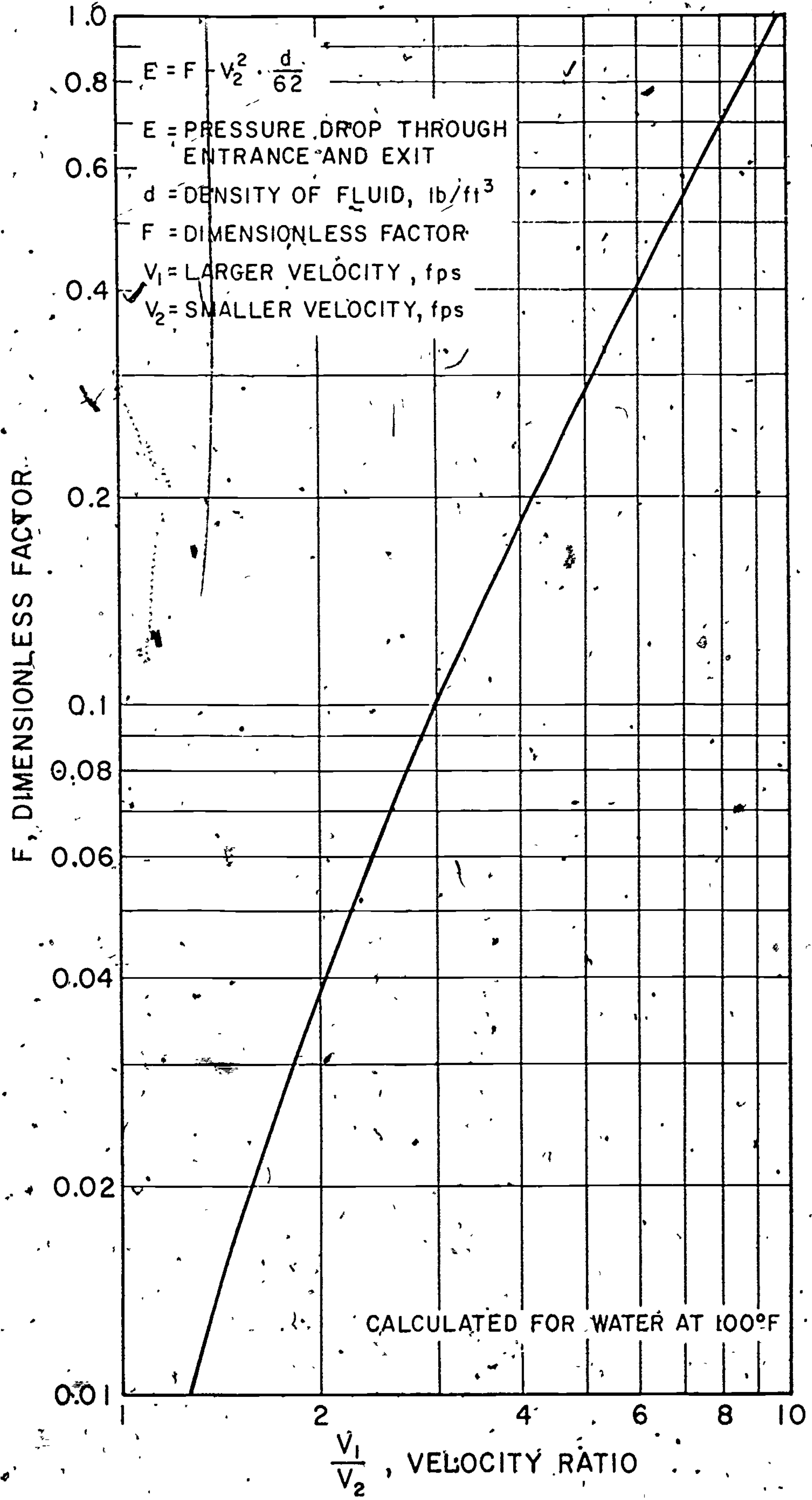


Figure 16-8. Entrance and Exit Pressure Drops.

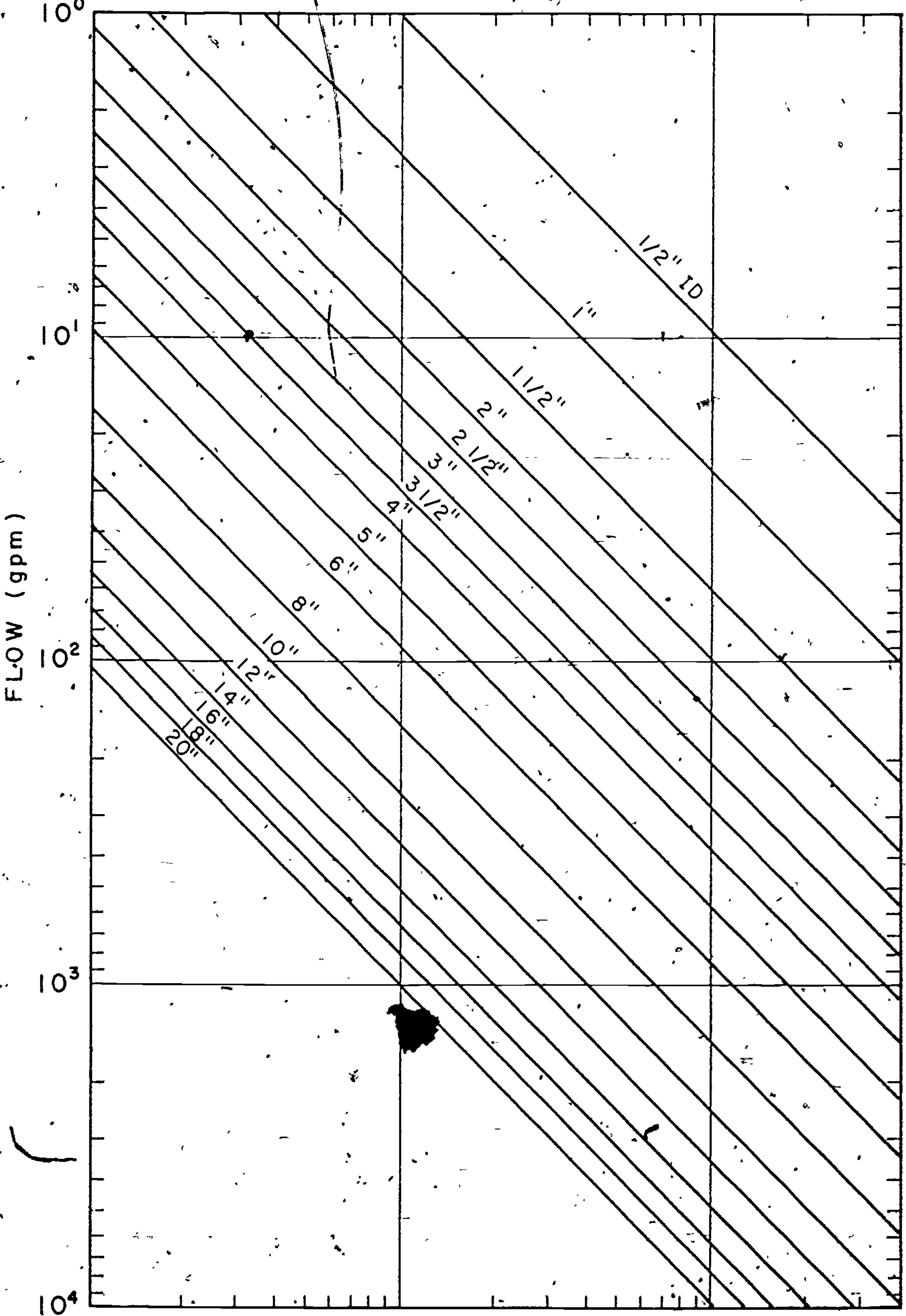


Figure 16-9. Nozzle and Bonnet Velocities.

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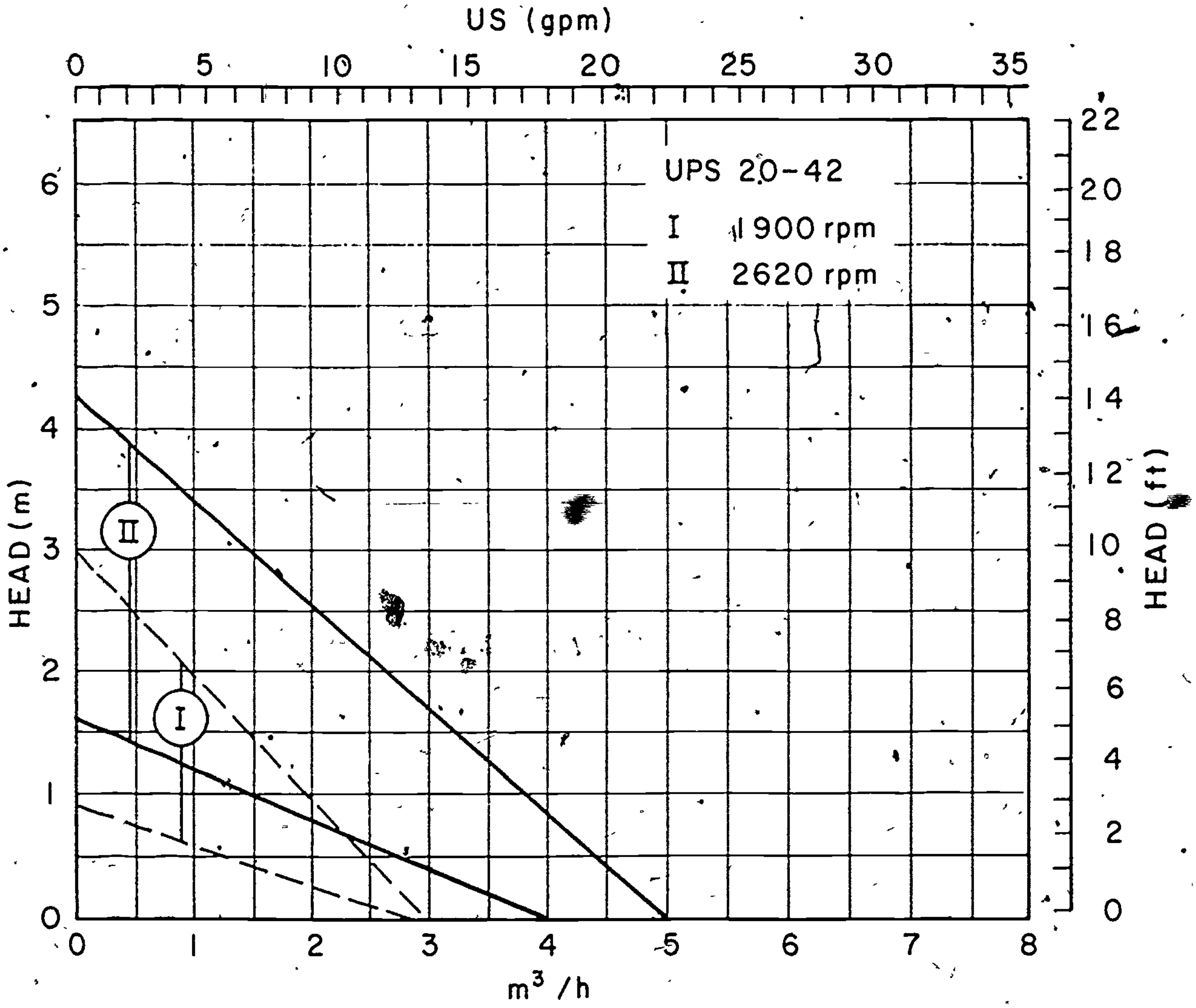

Figure 16-10. Typical Pump Performance Curves.

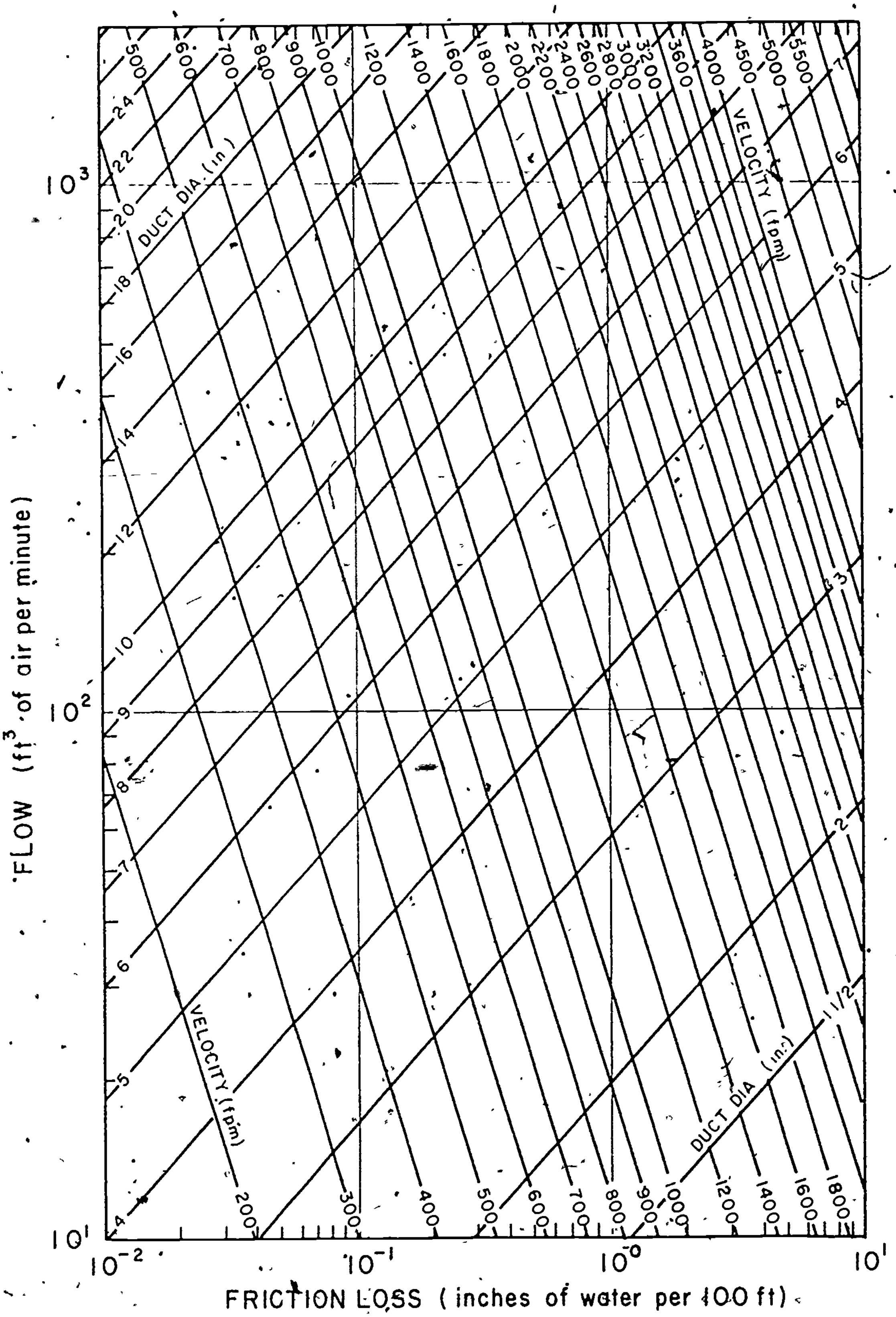

Figure 16-11. Friction Loss in a Straight Duct.

TRAINING COURSE IN
THE PRACTICAL ASPECTS OF
DESIGN OF SOLAR HEATING AND COOLING SYSTEMS
FOR
RESIDENTIAL BUILDINGS

MODULE 17

SOLAR COOLING SYSTEMS

SOLAR ENERGY APPLICATIONS LABORATORY
COLORADO STATE UNIVERSITY
FORT COLLINS, COLORADO

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GLOSSARY OF TERMS

absorbent	A liquid which combines chemically with a refrigerant
refrigerant	Working fluid in a refrigeration system
coefficient of performance	Ratio of heat removal rate to heat supply rate
ton of refrigeration	Heat removal at a rate of 12,000 Btu per hour

INTRODUCTION

Space cooling is withdrawal of heat from the air within a building enclosure to lower the temperature below that of the natural surroundings. Various solar cooling methods and systems are discussed in this module, and although methods using solar energy are of particular interest in this training course, other potential space cooling methods are briefly discussed.

TRAINEE-ORIENTED OBJECTIVE

The objective in this module is to present the basic principles and concepts of solar cooling methods for the purpose of designing a system using a solar cooling unit. In order to test whether this objective is met by the trainee, as a minimum level of accomplishment the trainee should be able to:

1. Describe the basic concepts of solar space cooling and determine the amount of annual energy consumption to operate the cooling unit.
2. Describe the operation cycles of the following experimental solar cooling systems: (a) lithium-bromide-water absorption cycle (b) an open-cycle liquid desiccant system.
3. Describe the operation of non-solar cooling units in solar heating systems.
4. Determine the economics of solar cooling systems for specific applications.

DEFINITION OF TERMS

The capacity of a refrigeration machine to cool room air is customarily referred in tons of refrigeration. A ton of refrigeration is the removal of heat at a rate of 12,000 Btu per hour. Another often-used term in connection with refrigeration equipment is coefficient of performance, COP. The COP expresses the effectiveness of a refrigeration system as the ratio of useful refrigeration effect to net energy supplied to the machine. The COP is determined by the simple equation below:

$$\text{COP} = \frac{\text{Heat energy removed}}{\text{Energy supplied from external sources}}$$

The COP of a mechanical vapor-compression refrigeration machine is characteristically about two, and can be as high as four. The COP of a lithium-bromide-water absorption refrigeration machine is about 0.8 and more often operates in the range from 0.6 to 0.7. A COP less than 1 means that more energy is supplied to the machine than heat energy removed from the room air. From the cooling capacity and COP the energy consumption rate by the machine to produce the cooling effect can be determined by dividing the heat removal rate by the COP. For example, the heat removal rate for a 3-ton absorption air cooler is 36,000 Btu per hour. With a COP of 0.6, the quantity of heat needed at the generator is 60,000 Btu per hour (36,000 ÷ 0.6).

CATEGORIES OF SPACE COOLING METHODS

There are three categories of space cooling units for residential buildings. They are:

1. Refrigeration
2. Evaporative Cooling
3. Radiative Cooling

Solar energy is directly useful in refrigeration methods and some evaporative cooling units. Simple evaporative cooling systems and radiative cooling are indirectly related to solar energy in that they are dependent upon climatic factors.

REFRIGERATION METHODS

Refrigeration systems effect cooling by removing heat from the air as it comes in contact with a cold, refrigerated surface. Conventional vapor-compression systems using electric motors as well as absorption vapor-compression systems using gas fuel heat are potentially convertible to systems using solar energy. Of many possible systems available, only the absorption systems appear to be useable from an economic view in the near-term (next five years), and of the various types of absorption machines possible, the lithium-bromide-water unit is currently (1977) commercially available for residential space-cooling applications.

ABSORPTION REFRIGERATION

An absorption refrigeration machine is basically a vapor-compression machine that accomplishes cooling by expansion of a liquid refrigerant under reduced pressure and temperature, similar in principle to an ordinary electrically operated vapor-compression air conditioner. Instead of a refrigerant like Freon in a conventional air conditioner, inorganic refrigerants such as water or ammonia are used in an absorption machine together with an absorbent. An absorbent is a liquid which combines chemically with the refrigerant and releases the heat from the fluid mixture in the combination process. Two units are described in this module, a lithium-bromide-water unit where water is the refrigerant and the lithium-bromide is the absorbent, and an ammonia-water unit where ammonia is the refrigerant and water is the absorbent.

In a mechanical vapor-compression system the compressor is driven by an electric motor; thus, the energy input is electricity. In an absorption vapor-compression system there is no compressor. Instead, there is a generator where energy input in the form of heat is used to drive the cooling machine. The quantity of heat energy needed by an absorption machine is greater than the amount of electrical energy (heat equivalent) needed to operate an electro-mechanical air conditioner to produce the same cooling capacity.

Lithium-Bromide-Water Absorption Cooler

There are two types of lithium-bromide-water absorption coolers. One type cools air and the other type cools water which contacts the cooling coils. The first type is called an air chiller; the second, a water chiller.

The principle of operation of a lithium-bromide-water system is described with the aid of Figure 17-1. Water is the refrigerant and

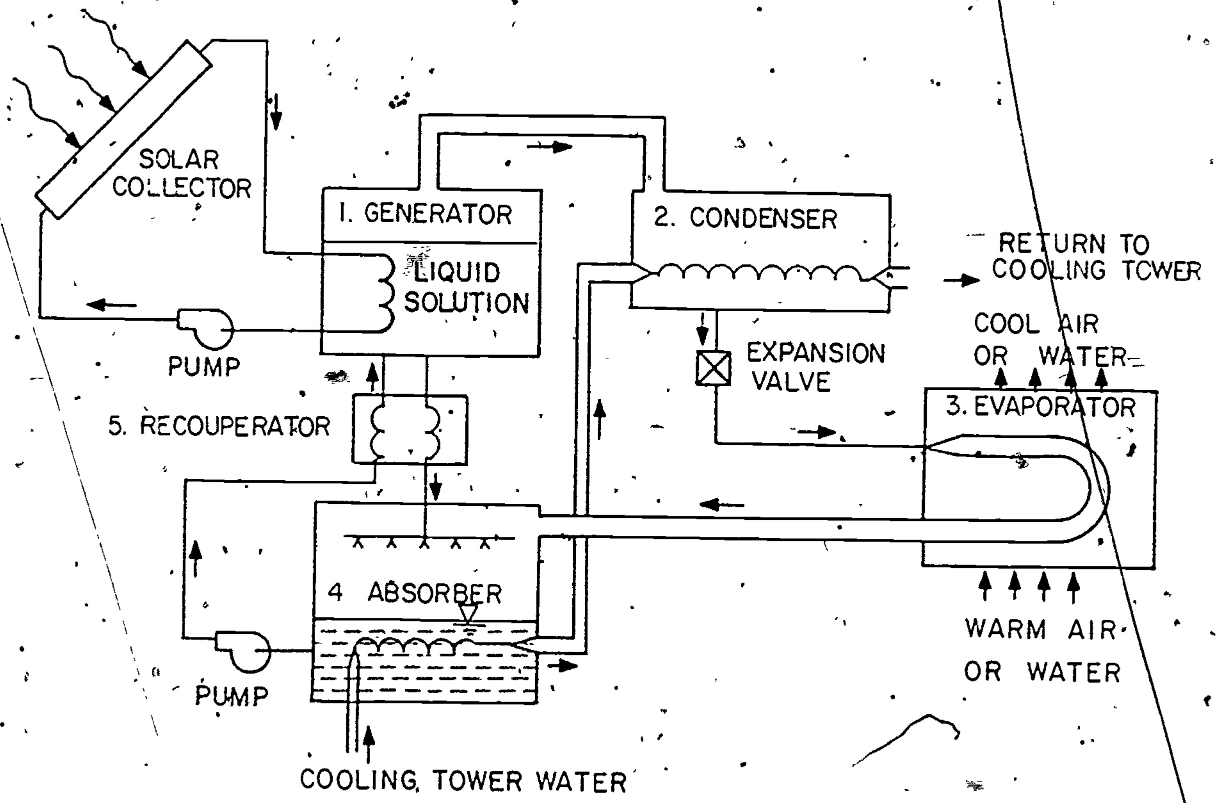
SOLAR COLLECTOR
1. GENERATOR
LIQUID SOLUTION
2. CONDENSER
RETURN TO COOLING TOWER
PUMP
EXPANSION VALVE
COOL AIR OR WATER
5. RECOUPERATOR
3. EVAPORATOR
4 ABSORBER
PUMP
WARM AIR OR WATER
COOLING TOWER WATER

Figure 17-1. Absorption Air Conditioner -- Schematic Drawing

the absorbent is lithium-bromide. The cycle begins when water in the liquid mixture in the generator is boiled off and superheated with solar energy at a temperature between 170 and 210 °F. The superheated water vapor passes from the generator to the condenser where it is cooled to about 100 °F by the cooling water from an outdoor cooling tower. The vapor condenses to a liquid and is then revaporized through an expansion valve which cools the vapor-liquid mixture to a temperature of 40 °F in the evaporator coils. The heat in the room air or water which is brought in contact with the evaporator is removed by the cooled refrigerant in the evaporator. The refrigerant then passes to the absorber where it recombines with the concentrated lithium-bromide solution from the generator at a temperature of about 100 °F. In this recombination process, heat is released, and the heat is removed by the cooling water from the cooling tower. The dilute solution of lithium-bromide and water in the absorber flows by gravity, or is pumped back, to the generator and the cycle is repeated. The recouperator in the diagram is a heat exchanger to make the system thermodynamically more efficient.

Temperature Restrictions -- The operating temperature range of the hot water supplied to the generator of a solar-operated lithium-bromide-water absorption refrigeration machine is restricted from 170 °F to 210 °F. The water temperature to the generator must be sufficiently high to boil the water from the solution in the generator. The temperature must be at least 170 °F. The upper temperature is limited to 210 °F because the hot water to the generator in a solar system is provided from storage and the temperature in storage will be less than the boiling temperature of water at atmospheric pressure. Another limitation is the temperature of the concentrated lithium-bromide solution which flows from the generator to the absorber through the recouperator. If the temperature is too low in the recouperator, and the concentration of the lithium-bromide-water solution is high, the lithium-bromide will solidify in the outlet tube leading from the recouperator to the absorber. Provided the temperature in the generator is between 170 °F to 210 °F, the unit will operate satisfactorily.

System Components -- The absorption cooler should be situated close to the hot water storage tank to minimize heat loss from the pipelines connected to the tank. A schematic diagram of a flow chart for a water chiller in a solar system is shown in Figure 17-2. The hot water from the top of the storage tank is pumped through the generator by pump P-2 and returned to the bottom of the tank. It will be noted that the piping connection goes through the auxiliary boiler. When the temperature in the storage tank is insufficient to operate the absorption chiller, the auxiliary boiler is used to provide heat to the generator. When the auxiliary boiler is used, the three-way valve at the bottom of the auxiliary boiler circulates the return water only through the auxiliary boiler. In this way, auxiliary energy is not used to heat the storage tank. The pump size and head depend upon the flow rate and pressure loss through the system and are influenced by the size and length of pipe connecting the storage tank to the generator and the return pipe from the generator to the storage tank.

A wet cooling tower is needed with the absorption chiller to discharge the heat from the condenser and the absorber to the atmosphere.

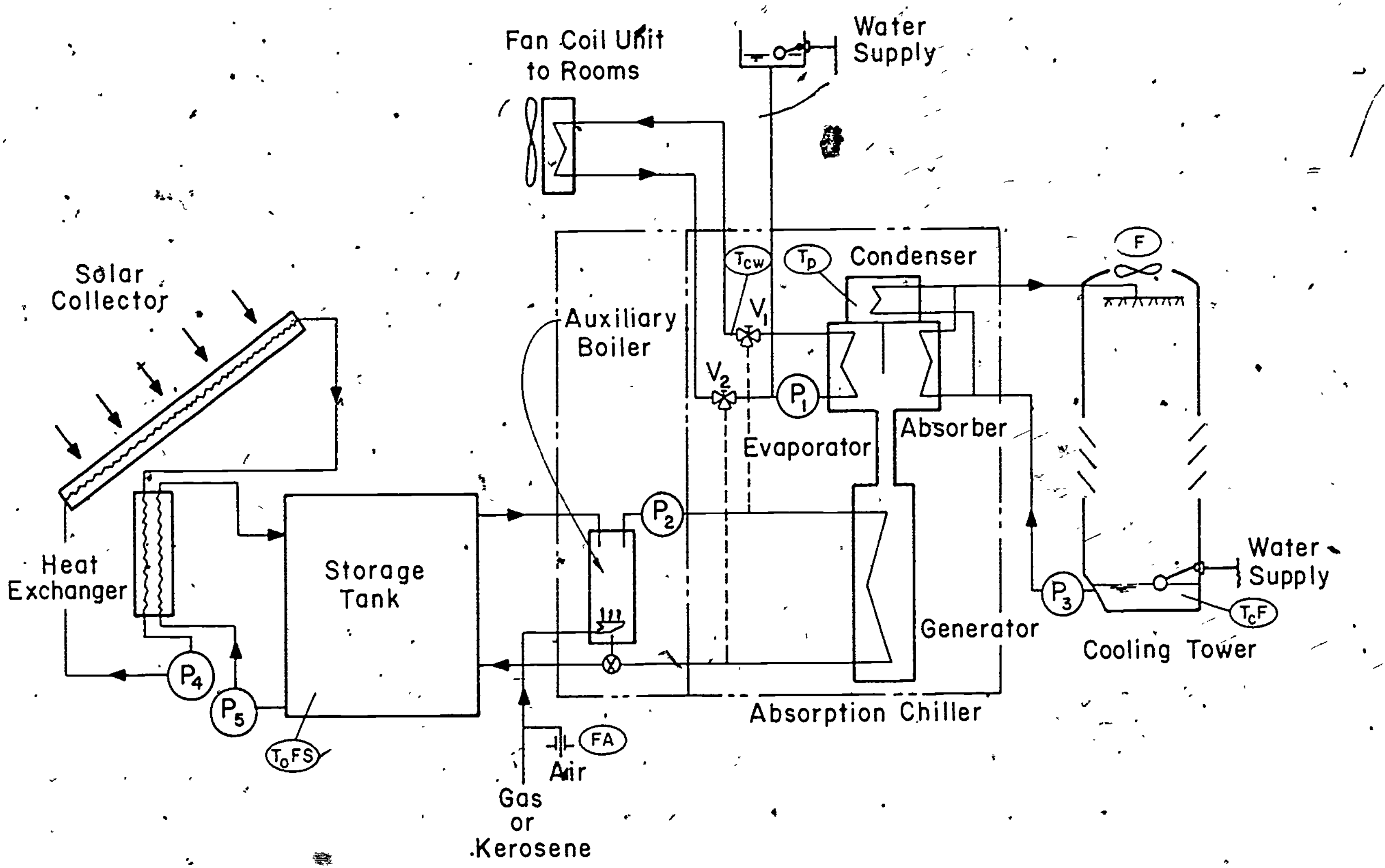
Fan Coil Unit to Rooms
Water Supply
T_{cw}
T_p
Condenser
F
Solar Collector
Auxiliary Boiler
V_1
V_2
P_1
Evaporator
Absorber
P_2
Heat Exchanger
Storage Tank
Generator
P_3
Water Supply
T_cF
Cooling Tower
P_4
P_5
T_oFS
FA
Air
Absorption Chiller
Gas or Kerosene

Figure 17-2. Flow Chart of a Water Chiller Operation

The size of the cooling tower needed depends upon the size of the absorption machine (cooling capacity) and the wet-bulb temperature of the ambient air. The temperature of the cooling water from the cooling tower will have a significant effect on the COP of the machine. For example, a drop in COP from 0.7 to 0.6 can be expected if the wet-bulb temperature increases from 75 °F to 85 °F. A pump labeled P-3 is needed to circulate the cooling water from the tower through the absorber and condenser of the absorption machine.

The chilled water from the evaporator is circulated to the fan-coil unit to cool the air in the rooms. The fan-coil unit may be a central unit for the entire building, or individualized units may be used in different zones within the building.

Water-Ammonia Absorption Cooler

In a water-ammonia absorption system, ammonia is the refrigerant and water is the absorbent. The system of Figure 17-1 must be modified slightly by inserting a separator between the generator and the condenser. The separator is needed to prevent excessive amounts of water vapor carry-over with the ammonia into the cycle. The operating pressure is greater than atmospheric, which is an advantage over the lithium-bromide system; however, operating temperatures are greater, which is a disadvantage. The temperatures are not so high as to be unattainable with solar collectors, especially if evacuated or concentrating collectors are available. As yet, solar energy operated ammonia-water absorption systems are not available. There are, however, gas-fired units available in larger sizes for large buildings.

HEAT PUMP

A heat pump is a device to absorb the heat from inside a building and reject it to the outside air, thus cooling a building. It can also be used to absorb heat from a source external to the building and reject it inside the building at a higher temperature level to heat the building.

A typical heat pump system is shown in the sketch of Figure 17-3 in the cooling mode. It is a mechanical vapor-compression system with a compressor, condenser, expansion valve and an evaporator. Warm indoor air is cooled at the evaporator and the cool air is redistributed through the building. The warm refrigerant vapor is compressed and heat is rejected through the condenser coils outside the building. As a refrigeration machine, the heat pump is not a solar energy related device. However, the same machine can be used for heating which can use a solar energy source.

In the heating mode, the cycle is reversed. The indoor coils now become the condenser and the outdoor coil, the evaporator. In the heating mode, as shown in Figure 17-4, heat, which is supplied by solar-heated water or air, is drawn into the refrigerant through the outdoor coils. The refrigerant vapor is compressed and condensed to a liquid in the indoor coils, thus giving up the heat to the room air. The cooled liquid is vaporized and returns to the evaporator to complete the cycle.

An important factor in the successful application of heat pumps for heating is the availability of a dependable source of heat for the evaporator at a reasonably high temperature. If solar energy can be such a source of supply, then a heat pump in combination with a solar heat source is a potentially important system for heating.

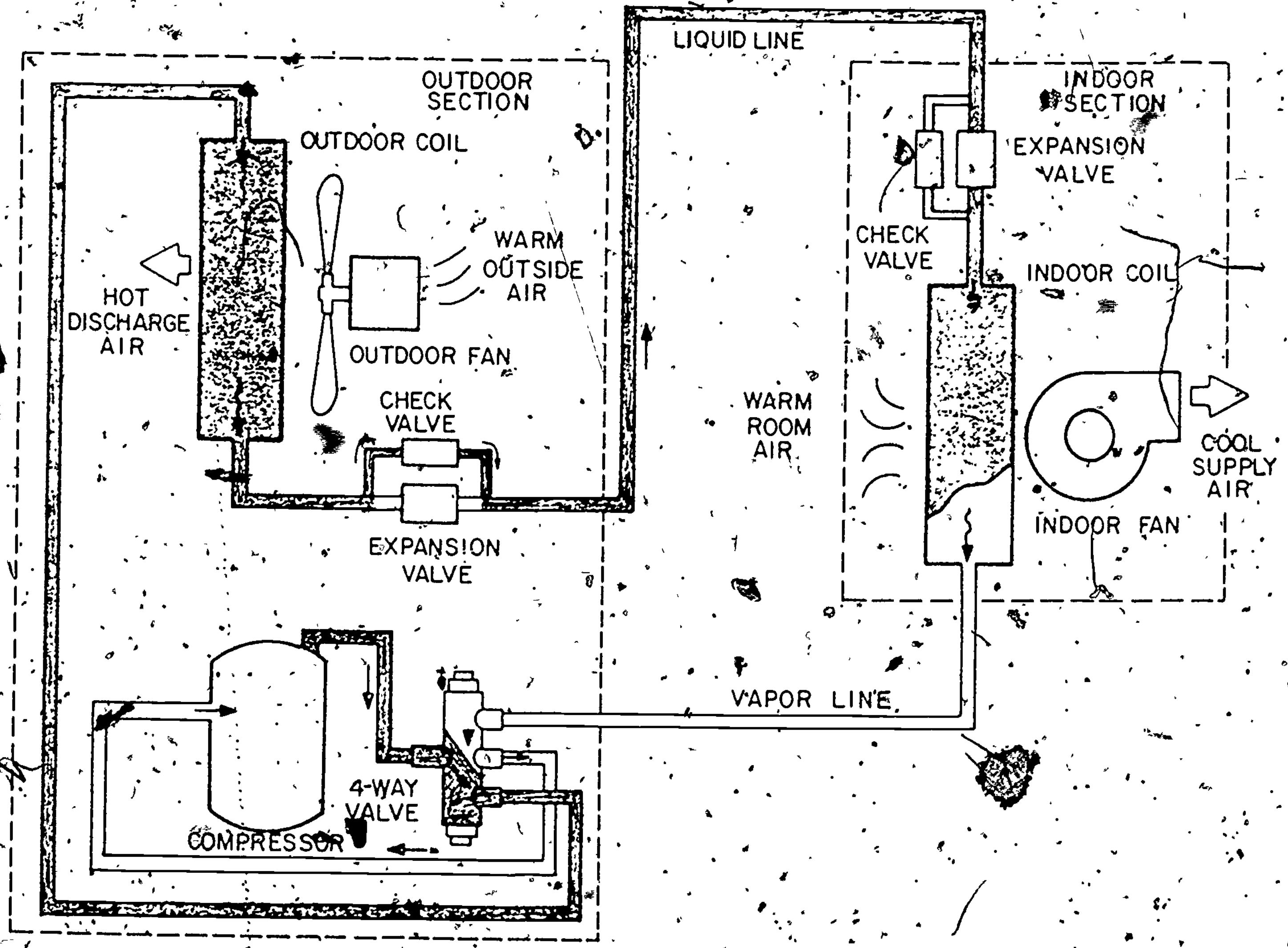
LIQUID LINE
OUTDOOR SECTION
OUTDOOR COIL
WARM OUTSIDE AIR
HOT DISCHARGE AIR
OUTDOOR FAN
CHECK VALVE
EXPANSION VALVE
INDOOR SECTION
EXPANSION VALVE
CHECK VALVE
INDOOR COIL
WARM ROOM AIR
COOL SUPPLY AIR
INDOOR FAN
VAPOR LINE
4-WAY VALVE
COMPRESSOR

Figure 17-3. Heat Pump in a Cooling Mode

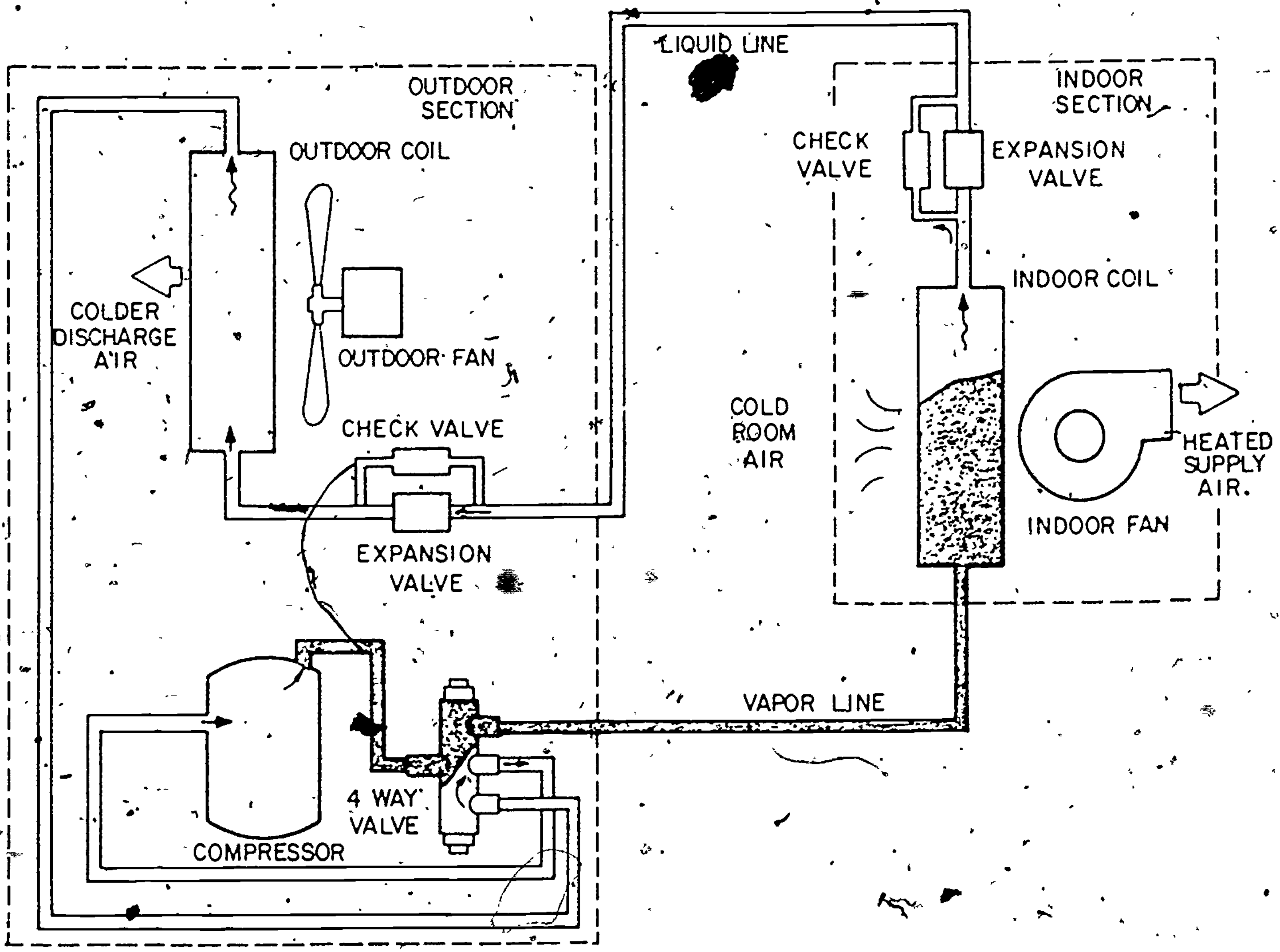
LIQUID LINE
OUTDOOR SECTION
INDOOR SECTION
OUTDOOR COIL
CHECK VALVE
EXPANSION VALVE
COLDER DISCHARGE AIR
INDOOR COIL
OUTDOOR FAN
CHECK VALVE
COLD ROOM AIR
HEATED SUPPLY AIR.
INDOOR FAN
EXPANSION VALVE
VAPOR LINE
4 WAY VALVE
COMPRESSOR

Figure 17-4. Heat Pump in a Heating Mode.

SOLAR RANKINE-CYCLE ENGINE

Instead of driving the compressor of a vapor-compression refrigeration machine with an electric motor, an alternative source of power for the compressor is a solar powered engine. Solar heat can be used to vaporize an organic fluid to drive a turbine. The turbine is coupled to a compressor of the refrigeration machine. A schematic drawing of a simplified system is shown in Figure 17-5.

Heat is supplied to the boiler by a solar collector. The fluid in the boiler is vaporized and the vapor drives the blades of the turbine. The rotating shaft of the turbine then drives a compressor for the vapor-compression refrigeration machine which produces the desired cooling effect. The vapor from the turbine is changed to a liquid in the condenser and is pumped back to the boiler. The regenerator is a heat exchanger to recover some of the heat from the vapor ejected from the turbine. This machine is still in the experimental stage and is not available as an operational unit for cooling of residential buildings.

EVAPORATIVE COOLING

EVAPORATIVE COOLING THROUGH ROCK BED

A simple evaporator cooler can be used to cool warm air by passing the air through an air washer. Depending upon the velocity of air and wet-bulb temperature, warm air may be evaporatively cooled to a desired dry-bulb temperature. As an example, outside air at 100 ^{o}F dry-bulb temperature and 70 ^{o}F wet-bulb temperature (relative humidity of 22 percent) can be cooled by an air washer to about 77 degrees. However, the relative humidity would be an uncomfortable 71 percent. Strictly

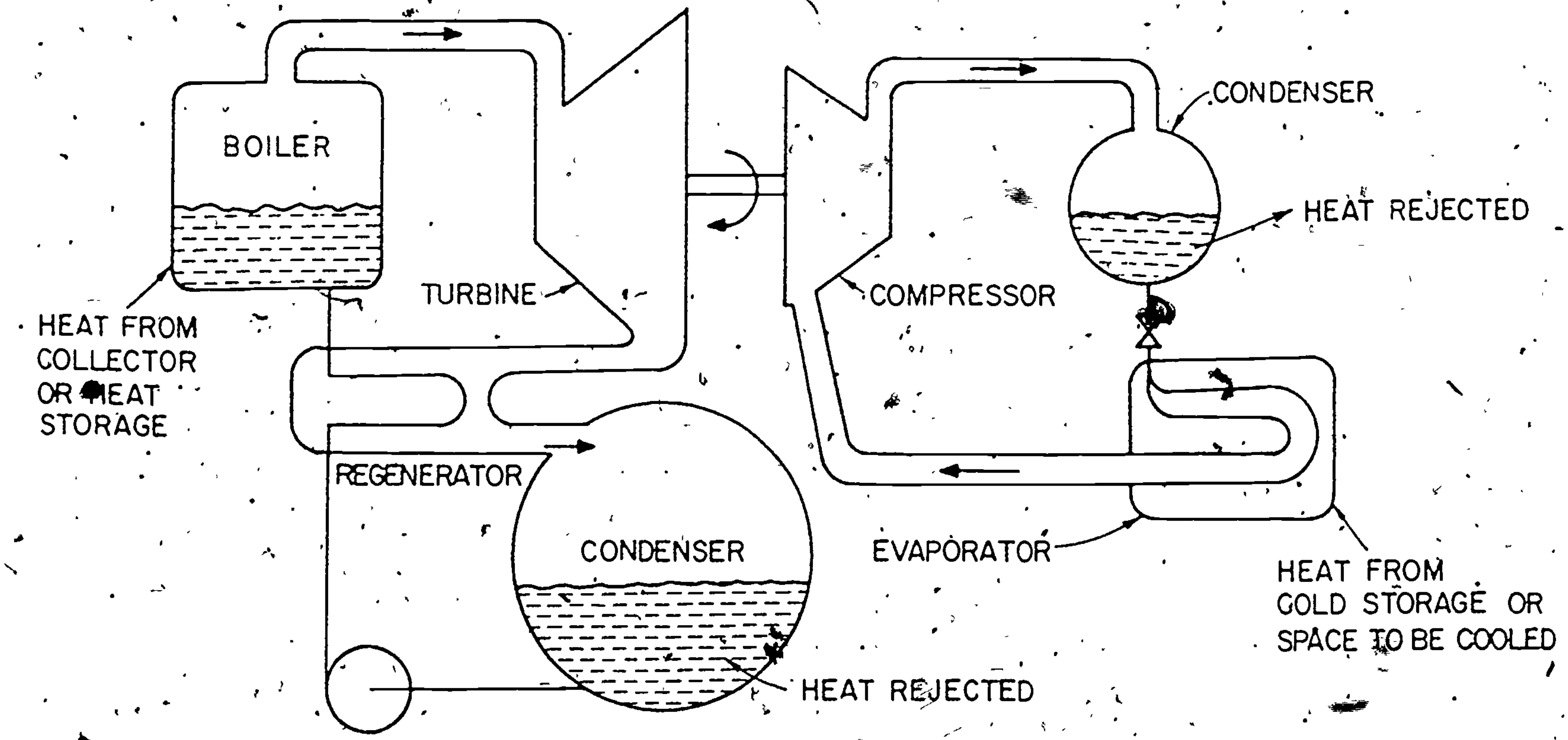
BOILER
TURBINE
HEAT FROM COLLECTOR OR HEAT STORAGE
REGENERATOR
CONDENSER
HEAT REJECTED
CONDENSER
HEAT REJECTED
COMPRESSOR
EVAPORATOR
HEAT FROM GOLD STORAGE OR SPACE TO BE COOLED

Figure 17-5. Rankine-Cycle Vapor-Compression System

speaking, evaporative cooling is not a solar system. However, there is the possibility of using the rock-bed storage of an air-heating solar system for storing "cool" air in the summer time as described below.

An evaporative cooler coupled with a rock-bed storage unit is shown in Figure 17-6; night air is evaporatively cooled and circulated through the rock bed to cool down the pebbles in the storage unit. During the day, warm air from the building can be cooled by passing the air through the cool pebble bed.

Using the design guidelines given for sizing the storage and blower of a solar air heating system, and assuming the collector area is 700 square feet, the rock-bed storage volume will be 350 ft.^{3} (minimum) to 700 ft.^{3} (maximum), and air flow rate will be about 1,400 cfm. Let it be assumed that the rock bed can be cooled down to 60 ^{o}F at night, and the desired temperature in the room is 78 ^{o}F (maximum) during the day. The rate of cooling provided by this system then is determined by (volume flow rate) x (air density) x (air heat capacity) x (temperature differential), or for this example the cooling rate is

$$\left(1400\ \frac{\text{ft.}^3}{\text{min}}\right)\left(.073\ \frac{\text{lbs.}}{\text{ft.}^3}\right)\left(.24\ \frac{\text{Btu}}{\text{lb}\ ^{o}\text{F}}\right)\left(78-60\ ^{o}\text{F}\right)\left(60\ \frac{\text{min}}{\text{hr}}\right) = 26,490\ \frac{\text{Btu}}{\text{hr}}$$

or 2.21 tons.

With a rock-bed storage volume of 350 ft.^{3}, the cooling capacity stored in the rocks is determined by

(volume of storage)(specific weight of rocks)(specific heat of rock) x (temperature difference) or,

$$(350\ \text{ft.}^3)(100\ \frac{\text{lbs}}{\text{ft.}^3})\ \times\ (0.21\ \frac{\text{Btu}}{\text{lb}\ ^{o}\text{F}})\ (78-60\ ^{o}\text{F}) = 132,300\ \text{Btu}$$

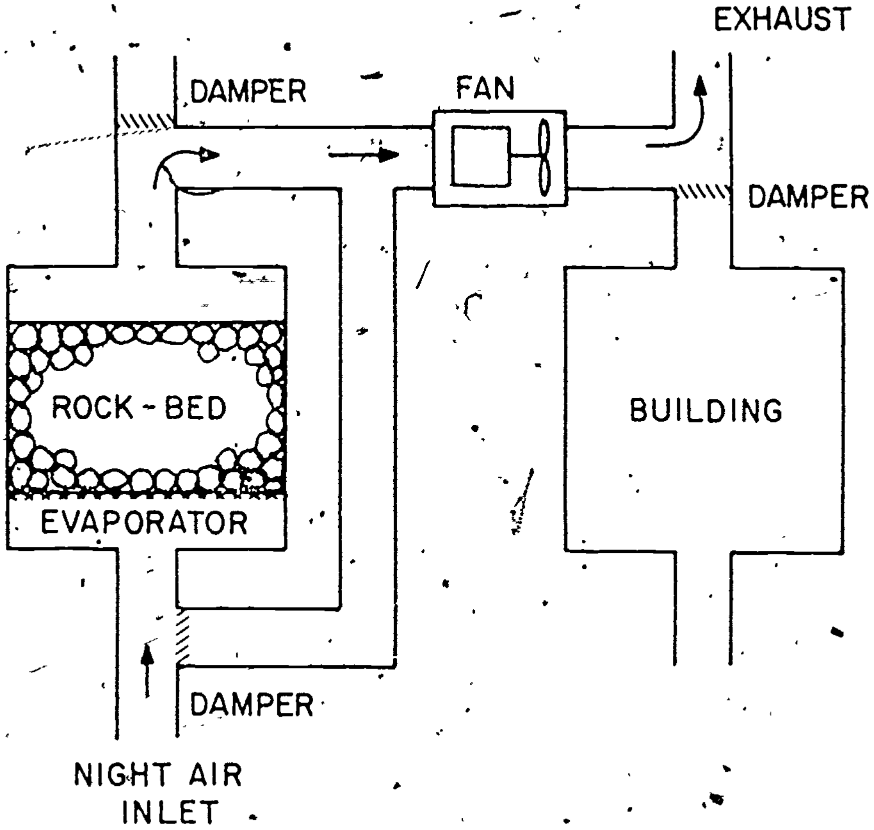
EXHAUST
DAMPER
FAN
DAMPER
ROCK - BED
EVAPORATOR
BUILDING
DAMPER
NIGHT AIR
INLET

Figure 17-6(a). Night Charging of Rock Bed

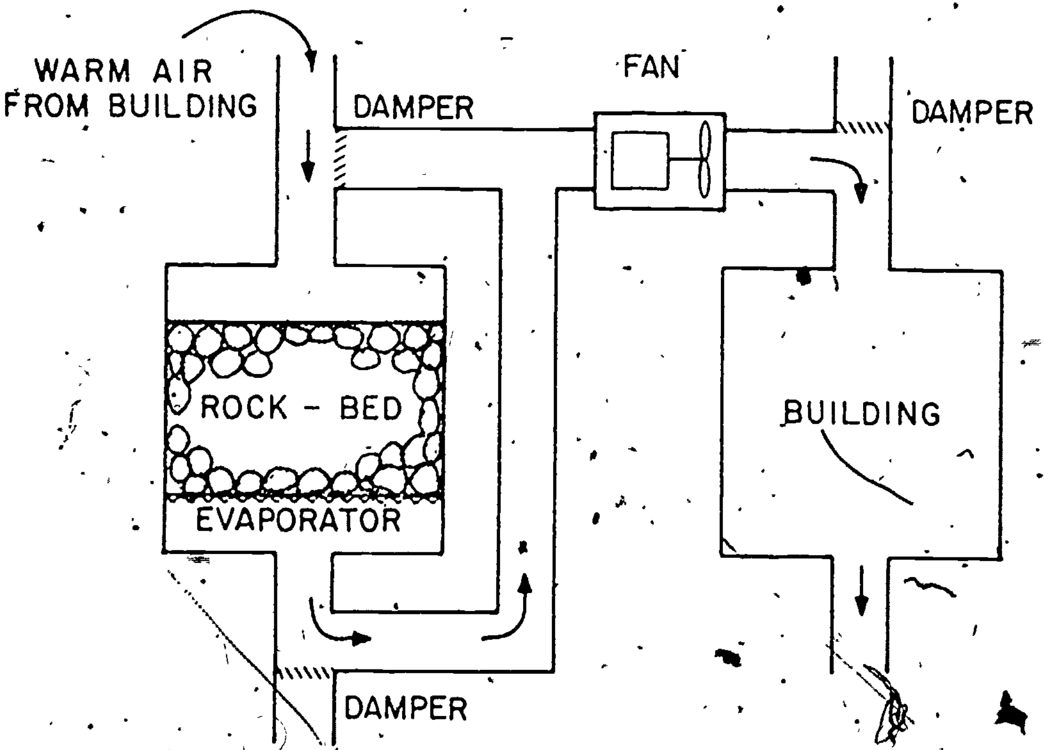
WARM AIR
FROM BUILDING
DAMPER
FAN
DAMPER
ROCK - BED
EVAPORATOR
BUILDING
DAMPER

Figure 17-6(b). Day Cooling of Building

At a cooling rate of 26,490 $\frac{Btu}{hr}$, there are 5 hours (132,300 ÷ 26,490) of cooling capability provided by 350 ft.^{3} of rock bed storage.

If the storage size is 700 ft.^{3}, the cooling is unchanged at 26,490 Btu per hour, or 2.21 tons, but the cooling capability is increased to 10 hours. When a solar air heating system with rock bed storage is considered for use in cooling, it is advantageous to install the maximum storage volume consistent with heating system design. On a unit collector area basis, storage size recommended is 0.5 to 1 cubic foot per square foot. Thus for a heating and cooling system, storage volume based on 1 cubic foot per square foot of collector is recommended to maximize the cooling capability of the system.

Evaporative cooling is restricted to arid and semi-arid regions with cool nights and low wet-bulb temperatures.

MUNTER'S ENVIRONMENTAL CONTROL

A Munter's Environmental Control (MEC) unit provides both cooling and heating. A solar assisted unit in a cooling mode is shown in Figure 17-7. The two essential parts of the MEC system are a drying wheel and a heat exchange wheel. The wheels operating in combination with a solar heater and gas burner for air drying and an evaporative cooler provide environmental control.

Hot moist air, at say, 90 °F dry bulb and 80 °F wet bulb, is drawn into the unit and dried nearly adiabatically to 180 °F D.B. and 80 °F W.B. by the drying wheel. The hot dry air is cooled by the slowly revolving heat exchange wheel to 75 °F/53 °F and further cooled adiabatically to 55 °F/53 °F by an evaporative cooler and distributed in the building.

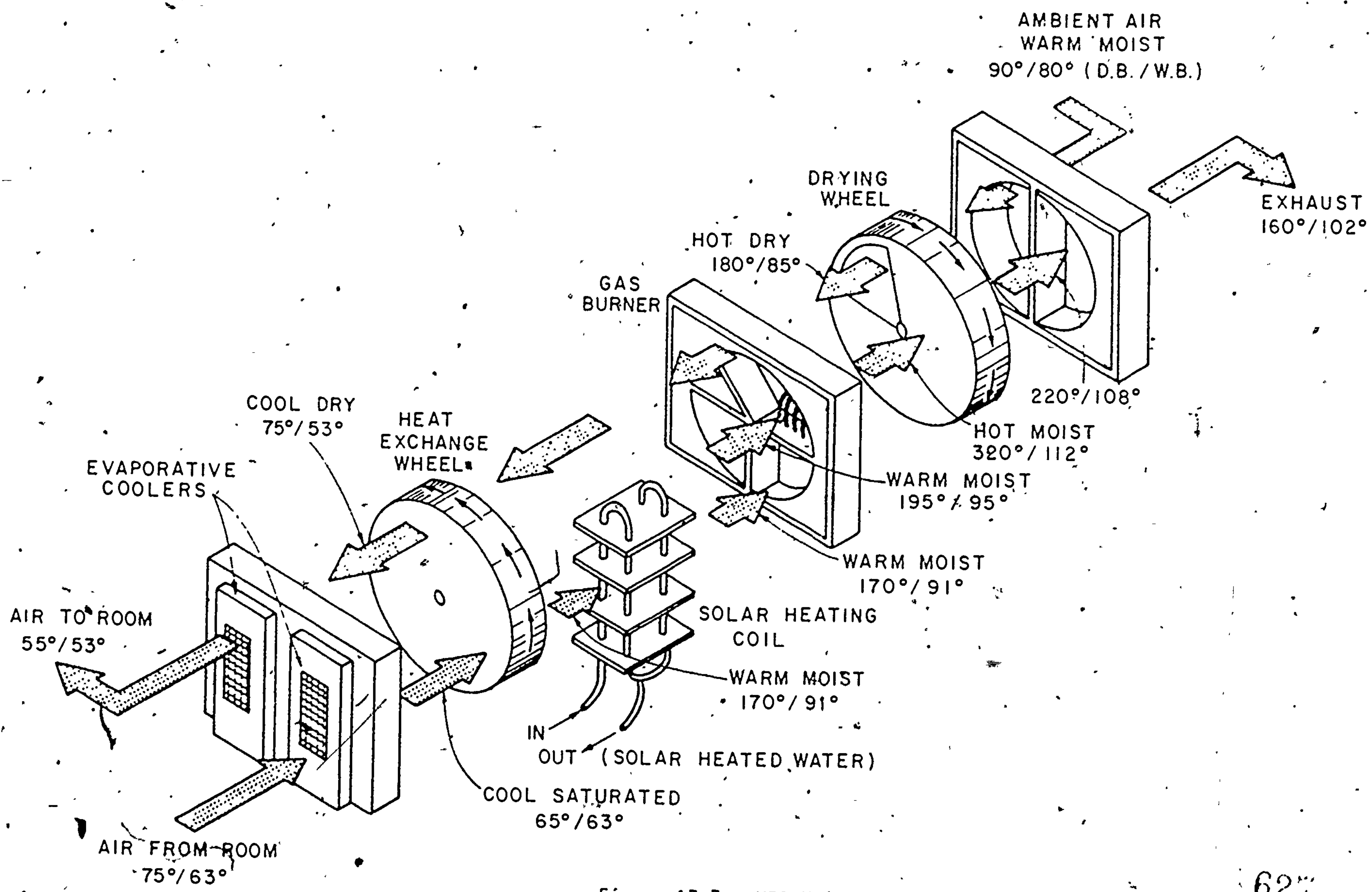
AMBIENT AIR
WARM MOIST
90°/80° (D.B./W.B.)
EXHAUST
160°/102°
DRYING WHEEL
HOT DRY
180°/85°
GAS BURNER
220°/108°
HOT MOIST
320°/112°
WARM MOIST
195°/95°
WARM MOIST
170°/91°
COOL DRY
75°/53°
HEAT EXCHANGE WHEEL
EVAPORATIVE COOLERS
SOLAR HEATING COIL
WARM MOIST
170°/91°
AIR TO ROOM
55°/53°
IN
OUT
(SOLAR HEATED WATER)
COOL SATURATED
65°/63°
AIR FROM ROOM
75°/63°

Figure 17-7. MEC Unit.

The drying wheel is regenerated by drawing the warm air from the room through the evaporative cooler where it is cooled from 75 °F/63 °F to 65 °F/63 °F. The cool air removes the heat from the heat exchange wheel, warming to 170 °F/91 °F. Part of this air is further heated by the solar heating coil and the gas burner to 320 °F. Part of the air from the heat exchange wheel is used for combustion in the gas burner, and the balance bypasses both the solar and gas heaters. The hot air regenerates the drying wheel by removing the moisture. The air from the drying wheel is combined with the unused air from the heat exchange wheel and is exhausted outdoors.

A 3-ton demonstration model has operated at a COP of about 0.3 with indoor temperature at 75 °F and 50-percent relative humidity. The contribution which solar energy makes to this system is limited by the temperature at the solar heating coil. To regenerate the drying wheel, a high temperature is desired, and assuming that a flat-plate collector is used on the building for heating purposes, the temperature rise across the solar heating coil is limited to about 25 °F. The balance in temperature rise, about 225 °F, is provided by the gas burner.

TRIETHYLENE GLYCOL OPEN-CYCLE DESICCANT SYSTEM

A system which provides cooling by dehumidification of the air is shown schematically in Figure 17-8. It is an open-cycle system because it does not require a hermetically sealed circulation system to contain a refrigerant. Moist room air is dehumidified and cooled by triethylene glycol as the air flows through the absorber. The dehumidified air passes through eliminators to remove the liquid glycol from the air and is further evaporatively cooled and redistributed to the

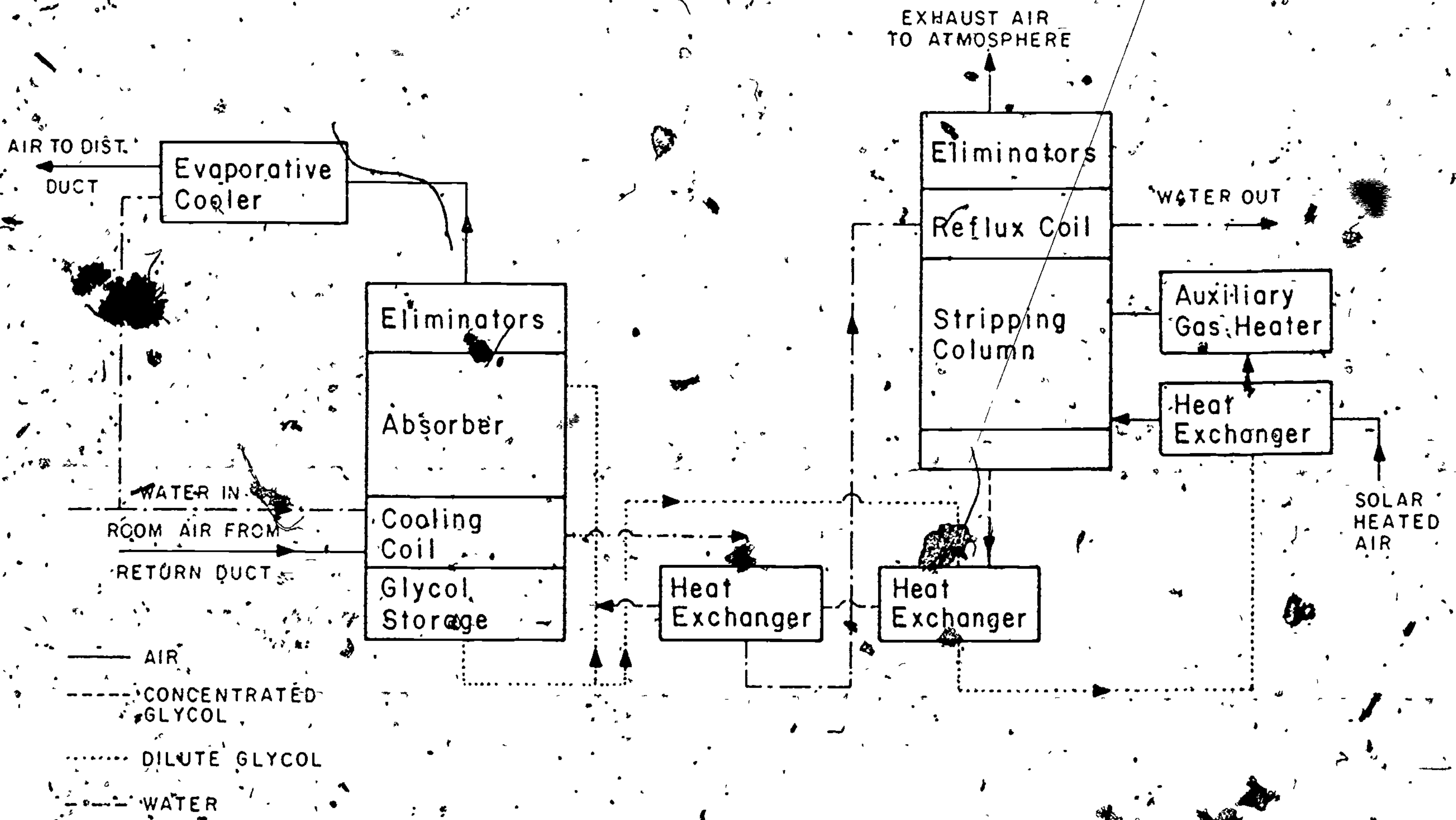
EXHAUST AIR TO ATMOSPHERE
AIR TO DIST. DUCT
Evaporative Cooler
Eliminators
Absorber
Cooling Coil
Glycol Storage
WATER IN
ROOM AIR FROM RETURN DUCT
Eliminators
Reflux Coil
WATER OUT
Stripping Column
Auxiliary Gas Heater
Heat Exchanger
SOLAR HEATED AIR
Heat Exchanger
Heat Exchanger
AIR
CONCENTRATED GLYCOL
DILUTE GLYCOL
WATER

Figure 17-8. Schematic of Triethylene Glycol (Liquid Desiccant) Open-Cycle Air Conditioning System

rooms. The liquid desiccant which passes through the absorber picks up moisture from the building air and becomes diluted. This dilute solution is regenerated by an outdoor unit on the right of Figure 17-8. The moisture is removed from the glycol solution in the stripping column, thus regenerating the triethylene glycol to a concentrated form which is returned to the absorber and recycled. At the stripping column the liquid mixture is sprayed into a stream of solar heated air. The heated air picks up the moisture from the glycol spray and is exhausted to the atmosphere. Liquid glycol droplets which are carried with the air stream are removed by the eliminators. If there is insufficient solar heat, then an auxiliary gas heater is used to heat the air stream. The triethylene glycol from the bottom of the stripping column returns to the absorber through heat exchangers to recover heat and make the system thermodynamically more efficient.

A wide range of operating solar heated air temperatures is possible with this system, from 84 °F to 180 °F. The higher the temperature, however, the higher will be the COP of the machine.

A liquid desiccant open-cycle system in large sizes, using conventional heat sources, is commercially available. Except for an experimental unit which was studied 25 years ago, this type of system has not been actively considered for space cooling of residential buildings.

RADIATIVE COOLING

The use of a flat-plate collector to cool water or air by night radiation in the cooling season has been suggested as a possible way to cool a building. In principle, radiation from the absorber surface of a flat-plate collector to the cold night sky could cool the absorber surface and hence also the water or air circulating through the collector. The difficulty with this method is that a good collector is a poor radiator; therefore, using the same collector which collects solar heat for the heating season to cool water or air in the cooling season is not practical.

There are two solar houses, one in California and the other in Arizona, that utilize night radiation to regulate the temperature rise in residential buildings. The buildings have a shallow water pond on the roof with sectionalized retracting insulating covers over the pond. The covers are retracted at night to cool the pond by evaporation and radiation to the night sky. The covers are closed during the day to prevent the pond from heating. In the winter, the insulating covers are retracted during sunny days to collect solar heat in the pond and closed at night to preserve the stored heat. At special locations in the country, this type of heating and cooling system could be used. However, in freezing climates there are obvious difficulties.

ECONOMICS OF SOLAR COOLING

At the present time, the only solar cooling system which has been satisfactorily and reliably operated is the lithium-bromide absorption machine. For economic considerations, suppose that the absorption machine costs approximately $2500. With a comparatively high load factor on the cooling unit, 750 sq. ft. of collectors should be able to provide 500,000 Btu of useful heat per average summer day, for the removal of approximately 300,000 Btu of heat from the building contents. This cooling rate is equivalent to about 25 ton-hours of cooling, that is, 3 tons for eight hours per day. On a seasonal basis, 100 days of cooling would result in a total of 2,500 ton-hours of cooling provided by the solar unit. Amortizing the costs of the air conditioner in 20 years at 8 percent, with all other costs such as the collectors, storage tank, pumps and ducts being included in the heating expense, the cost of cooling is approximately $250 per year. Dividing $250 by 2,500 ton-hours yields a cooling cost of 10 cents per ton-hour of cooling.

Comparison of the cost of a solar cooling system with the cost of conventional cooling can now be made. A vapor-compression cooling unit of 3 tons capacity, equivalent to the above, would require an investment of about $1500. At a cooling COP of about 2.0, 2,500 ton-hours of cooling require about 3750 kilowatt-hours of electric energy. At 3 cents per kilowatt-hour, the cost of electricity would be $112.50 per year. Adding $150 annualized cost of the equipment, approximately $262 per year, would provide 2,500 ton-hours of cooling. The cost per ton-hour is, therefore, approximately 10.5 cents. On the basis of the costs assumed in this simple comparison, solar cooling is competitive with conventional vapor-compression cooling.

For purposes of this illustration, electric power cost of 3 cents per kilowatt-hour was chosen. However, in many parts of the country, electricity-costs are much higher. It is easily seen, therefore, that with electricity costs of 3 to 5 cents in selected parts of the country, solar energy for cooling can be quite competitive. The critical assumptions made in this analysis were that the capital cost of the solar cooling unit was approximately $2500 and the balance of the solar system is economically justified by the heating demand. Obviously, if the capital cost of the cooling unit is higher, or part of the solar collectors, storage tank, and ancillary equipment is to be charged to the solar cooling system, then solar cooling is not competitive with conventional systems with electricity at 3 cents per kilowatt-hour or less. The cost of a lithium-bromide absorption water chiller from one manufacturer is reported to cost approximately $6,000. If the capital cost is indeed that large, solar cooling is not presently economical. It should be noted, however, that the high cost may be the result of low production quantity. It is likely that with mass-produced quantities of the solar cooling unit, the cost can be reduced substantially. Nevertheless, because large quantities of solar cooling units are not likely to be in demand for at least a few years, solar cooling is not presently an economic alternative. Within the next few years, however, the situation is expected to change.

TRAINING COURSE IN
THE PRACTICAL ASPECTS OF
DESIGN OF SOLAR HEATING AND COOLING SYSTEMS
FOR
RESIDENTIAL BUILDINGS

MODULE 18

AUTOMATED DESIGN TECHNIQUES

SOLAR ENERGY APPLICATIONS LABORATORY
COLORADO STATE UNIVERSITY
FORT COLLINS, COLORADO

INTRODUCTION

TRAINEE-ORIENTED OBJECTIVE

To become aware of the computer-aided design programs that are available for the design of systems that do not fit within the material which has been previously presented.

SUB-OBJECTIVES

1. To gain some familiarity with TRNSYS.
2. To gain some familiarity with SIMSHAC.
3. To learn about the Martin-Marietta/ERDA Program that is under development.

The design techniques that have been presented earlier during the course are based upon "standard" system arrangements. If one were to use these techniques for a situation that was quite different from the standard configurations, then the predicted system performance would likely be in error. As a specific example, if one wanted to use the Owens-Illinois Evacuated Tube Collector in a system, then some modification of the performance curves that are presented in Module 7 is required. As another example, suppose one wished to use a solar-augmented heat pump in a design. The design methods presented previously would not apply to the design of such a system. In order to design various "non-standard" systems, several computer-aided design programs have been developed. These programs simulate the performance of systems using numerical models of specific components that may be found in solar heating and cooling systems for buildings. The user of these programs can include the appropriate component models to assemble a complete solar heating and/or cooling system, and proceed to simulate the performance of this system with particular climatological conditions. With computer simulation of systems, the effect of different control strategies can be investigated to determine variations in an overall system performance due to changes in the size or type of any component of the system. Computer simulation programs are valuable analysis and design tools.

There are two primary programs in this category at the present time and a third is presently being developed. The first of these programs is TRNSYS, developed by faculty and students at the University of Wisconsin. TRNSYS has been widely disseminated and is extensively used. The program, a user's manual and documentation of the program, are available from the Solar Energy Laboratory at the University of Wisconsin in Madison. The program has been checked for internal consistencies (that is, verified), but has not yet been validated against actual performance data for a solar heated and/or cooled building. The model validation process is taking place as system performance data become available.

The second of these computer-aided design programs is SIMSHAC, an acronym for SIMulation of Solar Heating And Cooling Systems. This program was developed by Gearold R. Johnson and C. Byron Winn, and some of their associates at Colorado State University and has been used in the design of several solar systems that have been constructed. The program will be released as soon as the documentation is completed. SIMSHAC, as is the case with TRNSYS, has been verified so that users may be assured there are no programming errors. In addition, it has been validated against actual performance data of CSU Solar House I.

The third computer-aided design program, SOLCOST, developed by the Martin-Marietta Corporation, was scheduled to be released in 1977. SOLCOST differs from TRNSYS and SIMSHAC in that the user does not have any contact with computer inputs. The user simply completes a form that describes the structure, the solar system components, and the location, and a pre-processor provides information required by the computer design program. The advantage is that a person completely unfamiliar with computing can still make use of this automated design tool.

All three programs are similar in their basic approach. The details of SOLCOST, TRNSYS, and SIMSHAC can be obtained by writing to the developers of the programs.

The objective of automated design tools is to provide generalized solar energy system sizing and simulation programs which will be readily available and usable by all segments of the solar energy community. At Martin-Marietta, the approach for achieving this objective consists of expanding the Martin Interactive Thermal Analysis System (MITAS) into two solar programs:

1. SOLCOST - A simplified solar system design method for the non-engineer user. The program computes an optimum collector area and tilt angle from an analysis of life-cycle cost differences for a solar system versus a reference (conventional) HVAC system. If the user needs a heating and/or cooling load calculation, SOLCOST can compute it using a generalized thermal network contained in the program.

2. SOLSIM - This program performs system simulations for the engineering user who is familiar with thermal network methods. The plan is to provide MITAS input decks for a minimum of five typical solar systems. Starting with these basic systems, the user familiar with thermal network methods can readily modify the input to model his unique solar system.

SOLCOST is discussed in this course because it relates closely to the material that has been presented in previous modules. Refer to the SOLCOST Users Manual (supplementary handout), prepared by the Martin-Marietta Corporation.

TRAINING COURSE IN
THE PRACTICAL ASPECTS OF
DESIGN OF SOLAR HEATING AND COOLING SYSTEMS
FOR
RESIDENTIAL BUILDINGS

MODULE 19

SERVICE HOT WATER SYSTEMS

SOLAR ENERGY APPLICATIONS LABORATORY
COLORADO STATE UNIVERSITY
FORT COLLINS, COLORADO

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INTRODUCTION

The oldest and simplest domestic use of solar energy is for heating water. Solar hot water heaters were used in the United States at least 75 years ago, first in southern California and later in southern Florida. Although the use of solar water heaters in these regions declined during the last 40 years, use in Australia, Israel, and Japan has risen rapidly, particularly in the last 15 years.

In its simplest form, a solar water heater comprises a flat-plate water heating collector and an insulated storage tank positioned at a higher level than the collector. These components, connected to the cold water main and the hot water service piping in the dwelling, provide most of the hot water requirements in a sunny climate. Nearly all of the solar hot water systems used in the United States have been of this type.

OBJECTIVES

The objectives are to choose a particular arrangement suitable for a given location, size the system for a given collector type and hot water requirement, install the system, and be confident of satisfactory operation. From the contents of this module the trainee should be able to:

1. Identify the types of domestic hot water systems available,
2. Select a domestic hot water system for a particular location and application,
3. Integrate a domestic hot water system into a space heating system,

4. Install and put into operation a domestic hot water system.

Maintain a domestic hot water system.

TYPES AND CHARACTERISTICS OF SOLAR HOT WATER HEATERS

Most of the solar water heaters that have been experimentally and commercially used can be placed in two main groups:

1. Circulating types, involving the supply of solar heat to a fluid circulating through a collector and storage of hot water in a separate tank
2. Non-circulating types, involving the use of water containers that serve both as solar collector and storage.

The circulating group may be divided into the following types and sub-types:

1. Direct heating, single-fluid types in which the water is heated directly in the collector, by:
   a. Thermosiphon circulation between collector and storage
   b. Pumped circulation between collector and storage
2. Indirect heating, dual-fluid types in which a non-freezing medium is circulated through the collector for subsequent heat exchange with water, when:
   a. Heat transfer medium is a non-freezing liquid
   b. Heat transfer medium is air.

DIRECT HEATING, THERMOSIPHON CIRCULATING TYPE

The most common type of solar water heater, used almost exclusively in non-freezing climates, is shown in Figure 19-1. The collector, usually

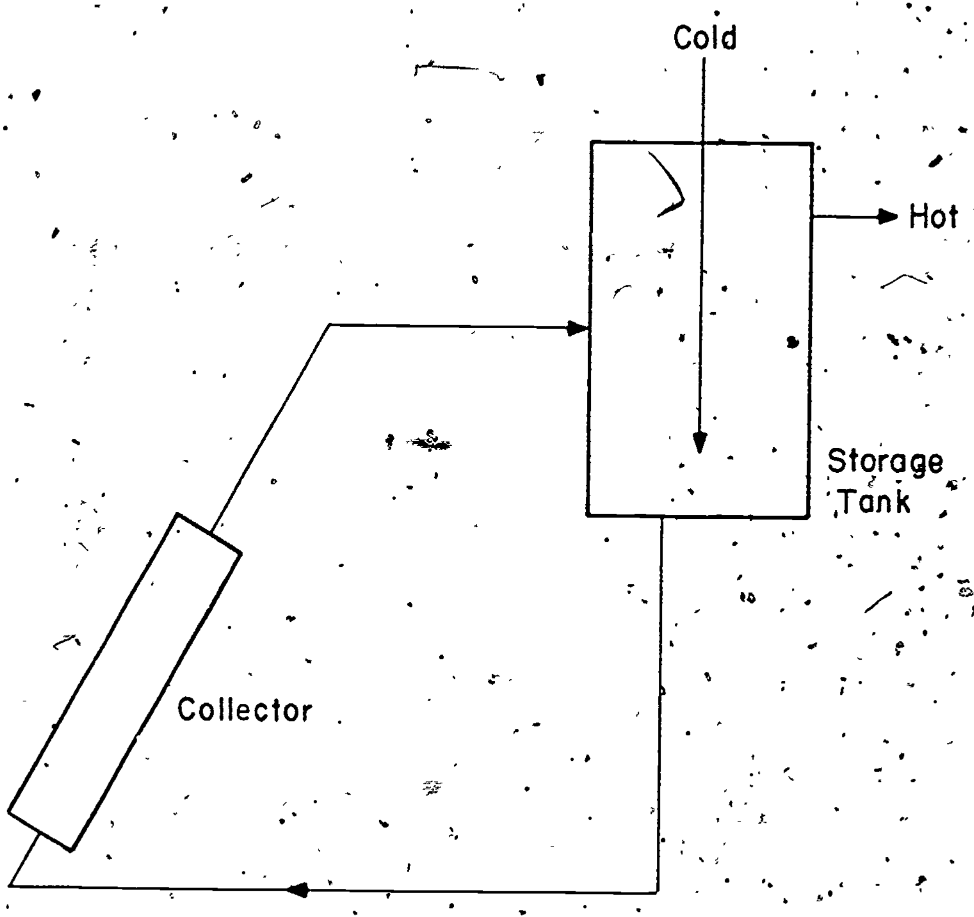

Figure 19-1. Direct Heating Thermosiphon Circulation Type of Solar Water Heater

single glazed, may vary in size from about 30 square feet to 80 square feet, whereas the insulated storage tank is commonly in the range of 40 to 80 gallons capacity. The hot water requirements of a family of four persons can usually be met by a system in the middle of this size range, in a sunny climate. Operation at supply line pressure can be provided if the system is so designed. With a float valve in the storage tank or in an elevated head tank, unpressurized operation can be utilized if the system is not designed for pressure. In the latter case, gravity flow from the hot water tank to hot water faucets would have to be accepted, or an automatic pump would have to be provided in the hot water line to supply pressure service. Plumbing systems and fixtures in the United States normally require the pressurized system.

Location of the tank higher than the top of the collector permits circulation of water from the bottom of the tank through the collector and back to the top of the tank. The density difference between cold and hot water produces the circulating flow. Temperature stratification in the storage tank permits operation of the collector under most favorable conditions, water at the lowest available temperature being supplied to the collector and the highest available temperature being provided to service. Circulation occurs only when solar energy is being received, so the system is self-controlling. The higher the radiation level, the greater the heating and the more rapid the circulating rate will be. In a typical collector under a full sun, a temperature rise of 15°F to 20°F is commonly realized in a single pass through the collector.

To prevent reserve circulation and cooling of stored water when no solar energy is being received, the bottom of the tank should be located above the top header of the collector. If the collector is on a house roof, the tank may also be on the roof or in the attic space beneath a sloping roof.

Although seldom used in cold climates, the thermosiphon type of solar water heater (storage tank above collector) can be protected from freezing by draining the collector. To avoid draining the storage tank also, thermostatically actuated valves in the lines between collector and storage tank must close when freezing threatens, a collector drain valve must open, and a collector vent valve must also open. The collector will then drain and air will enter the collector tubes. Water in the storage tank, either inside the heated space or sufficiently well insulated to avoid freezing, does not enter the collector during the period when

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sub-freezing temperatures threaten. Resumption of operation requires closure of the drain and vent valves and opening of the valves in the circulating line. The possibility of control failure or valve malfunction makes this complex system unattractive in freezing climates.

DIRECT HEATING, PUMP CIRCULATION TYPES

If placement of the storage tank above the collector is inconvenient or impossible, the tank may be located below the collector and a small pump used for circulating water between collector and storage tank. This arrangement is usually more practical than the thermosiphon type in the United States, because the collector would often be located on the roof with a storage tank in the basement. Instead of thermosiphon circulation when the sun shines, a temperature sensor actuates a small pump which circulates water through the collector-storage loop. A schematic arrangement is shown in Figure 19-2. To obtain maximum utilization of solar energy,

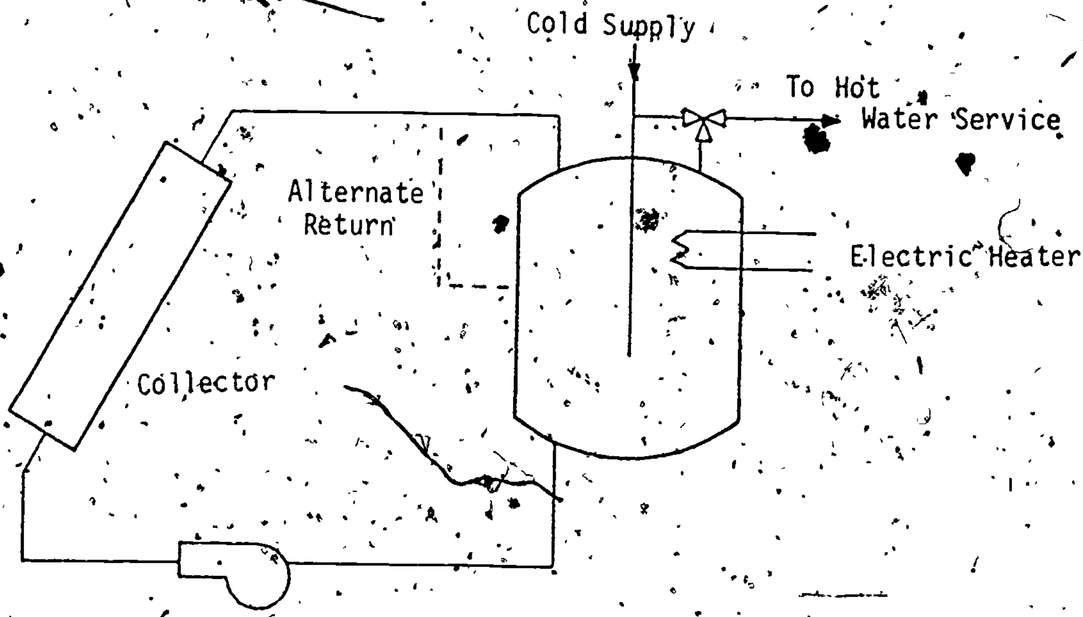


Figure 19-2. Direct Heating, Pump Circulation Type of Solar Water Heater

control is based on the difference in water temperature at collector outlet and bottom of storage tank. Whenever this difference exceeds a preset number of degrees, say 10°F, the pump motor is actuated. The sensor at the collector outlet must be located close enough to the collector so that it is affected by collector temperature even when the pump is not running. Similarly, the sensor in the storage tank should be located in or near the bottom outlet from which the collector is supplied. When the temperature difference falls below the preset value, the pump is shut off and circulation ceases. To prevent reverse thermosiphon circulation and consequent water cooling when no solar energy is being received, a check valve should be located in the circulation line.

If hot water use is not sufficient to maintain storage tank temperature at normal levels (as during several days of non-use), boiling may occur in the collector. If a check valve or pressure-reducing valve prohibits back flow from the storage tank into the main, a relief valve must be provided in the collector-storage loop. The relief valve will permit the escape of steam and prevent damage to the system.

DIRECT HEATING, PUMP CIRCULATION, DRAINABLE TYPES

If the solar water heater described above is used in a cold climate, it may be protected from freeze damage by draining the collector when sub-freezing temperatures are encountered. Several methods can be used. Their common requirement, however, is reliability, even when electric power may not be available. One method is shown in Figure 19-3.

Drainage of the collector in freezing weather can be accomplished by automatic valves which provide water outflow to a drain (sewer) and the inflow of air to the collector. The control system can be arranged so that whenever the circulating pump is not in operation, these two

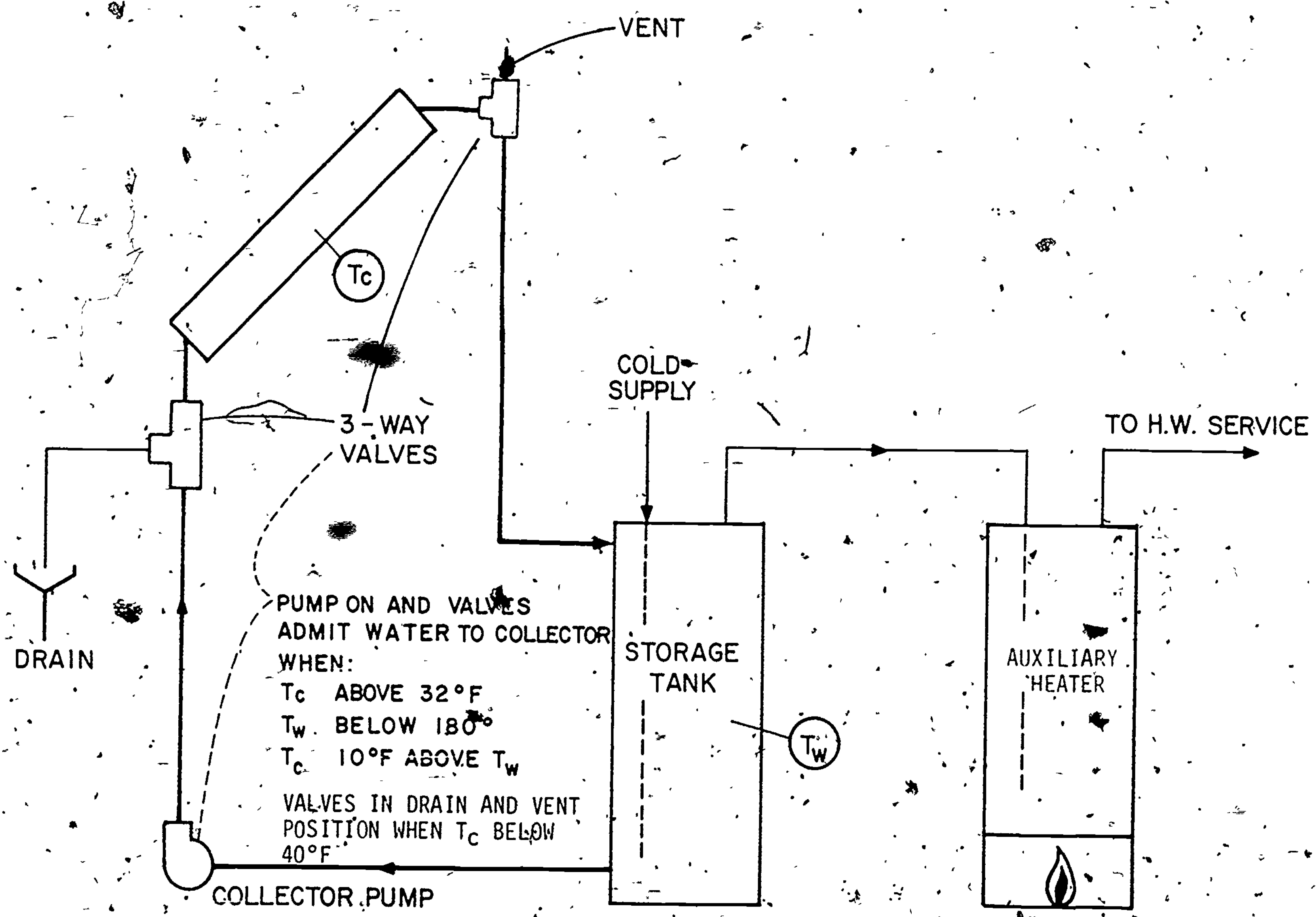

VENT
Tc
COLD SUPPLY
3-WAY VALVES
TO H.W. SERVICE
DRAIN
PUMP ON AND VALVES ADMIT WATER TO COLLECTOR WHEN:
Tc ABOVE 32°F
Tw BELOW 180°
Tc 10°F ABOVE Tw
VALVES IN DRAIN AND VENT POSITION WHEN Tc BELOW 40°F
STORAGE TANK
Tw
AUXILIARY HEATER
COLLECTOR PUMP

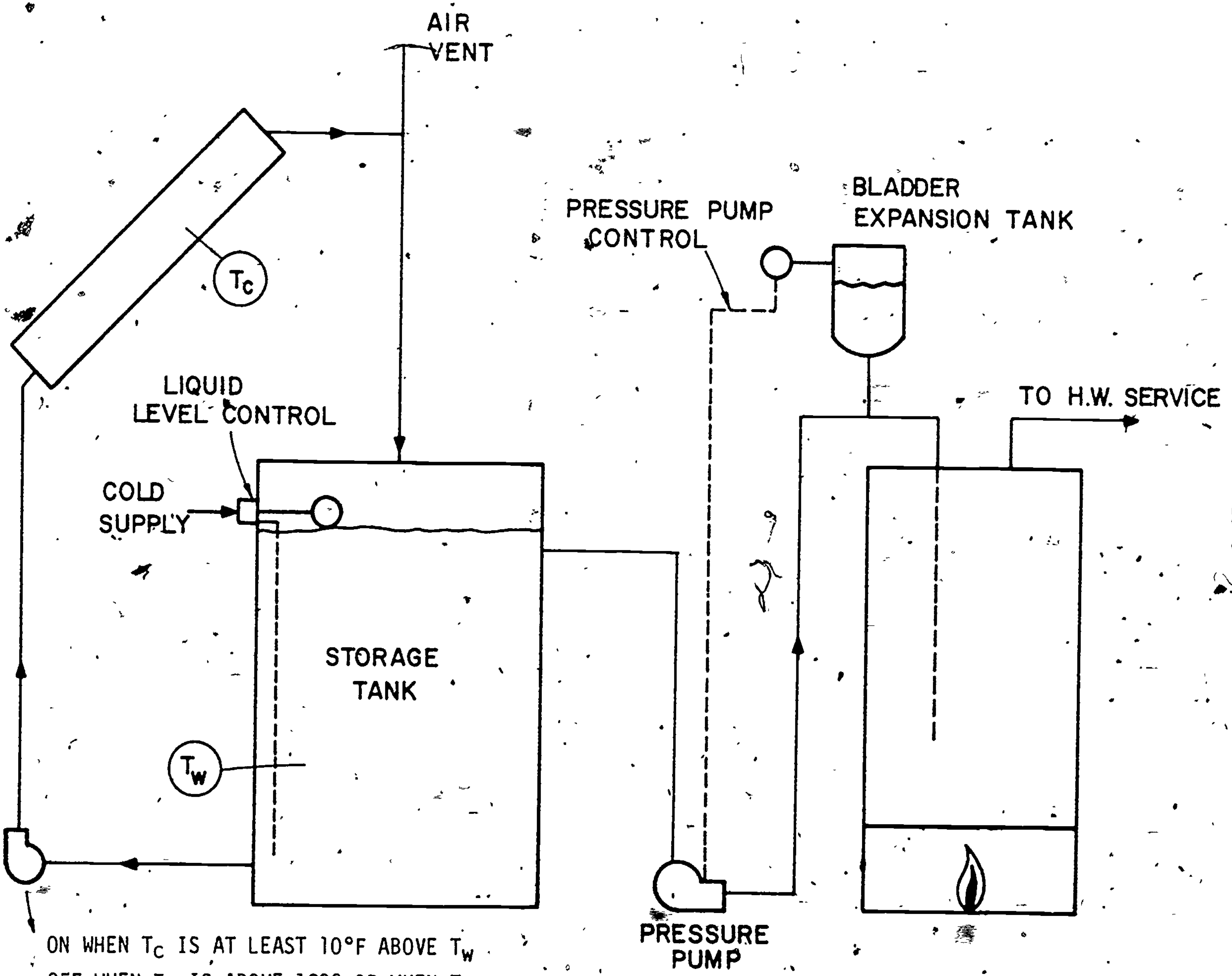
Figure 19-3. Solar Water Heater with Freeze Protection by Automatic Collector Drainage

valves are open. To assure maximum reliability, the valves should be mechanically driven to the drain position (by springs or other means), rather than electrically, so that in the event of a power failure, the collector can automatically drain.

The drainage system shown in Figure 19-3 is actuated by the temperature sensor, T_c, in the collector. When the sensor indicates a possibility of freezing, it can open the drainage and vent valves, thereby providing protection. The temperature sensor can be of the vapor pressure type, with capillary tube connections to mechanical valve actuators, or of the electrical type where the valves are held open by electrical means, automatically closing either when electrical failure occurs, or at low temperatures.

Another possibility for drainage of the collector is based on use of a non-pressurized collector and storage assembly as shown in Figure 19-4. A float valve in the storage tank controls the admission of cold water to the tank, and a pump in the hot water distribution system can furnish the necessary service pressure. With this design, the solar collector drains into the storage tank whenever the pump is not operating, as air enters the collector through a vent.

Start-up of any of the vented collector systems must permit the displacement of air from the collector. In either the line-pressure system or the unpressurized system, the entry of water into the collector (from the shut-off valve or pump) forces air from the collector tubes as long as the vent remains open. The vent valve design can be of a type which automatically passes air but shuts off when water reaches it.


AIR VENT
T_C
PRESSURE PUMP CONTROL
BLADDER EXPANSION TANK
LIQUID LEVEL CONTROL
COLD SUPPLY
TO H.W. SERVICE
STORAGE TANK
T_W
PRESSURE PUMP

ON WHEN T_C IS AT LEAST 10°F ABOVE T_W
OFF WHEN T_W IS ABOVE 180° OR WHEN T_C
IS NOT AT LEAST 10°F ABOVE T_W

Figure 19-4. Unpressurized Vented Solar Water Heater System

CIRCULATING TYPE, INDIRECT HEATING

As can be inferred from the above discussions of needs and means for collector drainage in freezing climates, costs and hazards are involved with those systems. The drainage requirement can be eliminated by the use of a non-freezing heat transfer medium in the solar collector, and a heat exchanger (inside the building) for transfer of heat from the solar heat collecting medium to the service water. The collector need never be drained, and there is no risk of freezing and damage. Corrosion rate in the wet collector tubes is also decreased when intermittent admission of oxygen is not required.

Liquid Transfer Media

Figure 19-5 illustrates a method for solar water heating with a liquid heat transfer medium in the solar collector. The most commonly used liquid is a solution of ethylene glycol (which is common automobile radiator antifreeze) in water. A pump circulates this unpressurized solution, as in the direct water heating system, and delivers the liquid to and through a liquid-to-liquid heat exchanger. Simultaneously, another pump circulates domestic water from the storage tank through the exchanger, back to storage. The control system is essentially the same as that in the design employing water in the collector directly. If the heat exchanger is located below the bottom of the storage tank, and if the pipe sizes and heat exchanger design are adequate, thermosiphon circulation of water through the heat exchanger can be used. A small expansion tank needs to be provided in the collector loop, preferably near the high point of the system, with a vent to the atmosphere.

To meet most code requirements, the heat exchanger must be of a design such that rupture or corrosion failure will not permit flow from

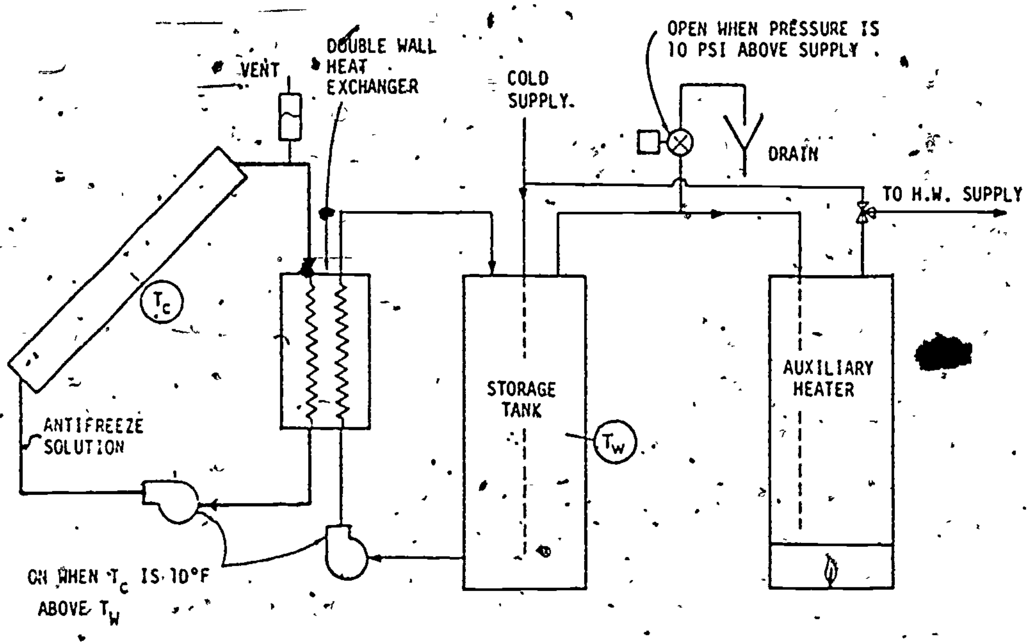

VENT
DOUBLE WALL HEAT EXCHANGER
COLD SUPPLY
OPEN WHEN PRESSURE IS 10 PSI ABOVE SUPPLY
DRAIN
TO H.W. SUPPLY
T_c
STORAGE TANK
T_w
AUXILIARY HEATER
ANTIFREEZE SOLUTION
ON WHEN T_c IS 10°F ABOVE T_W

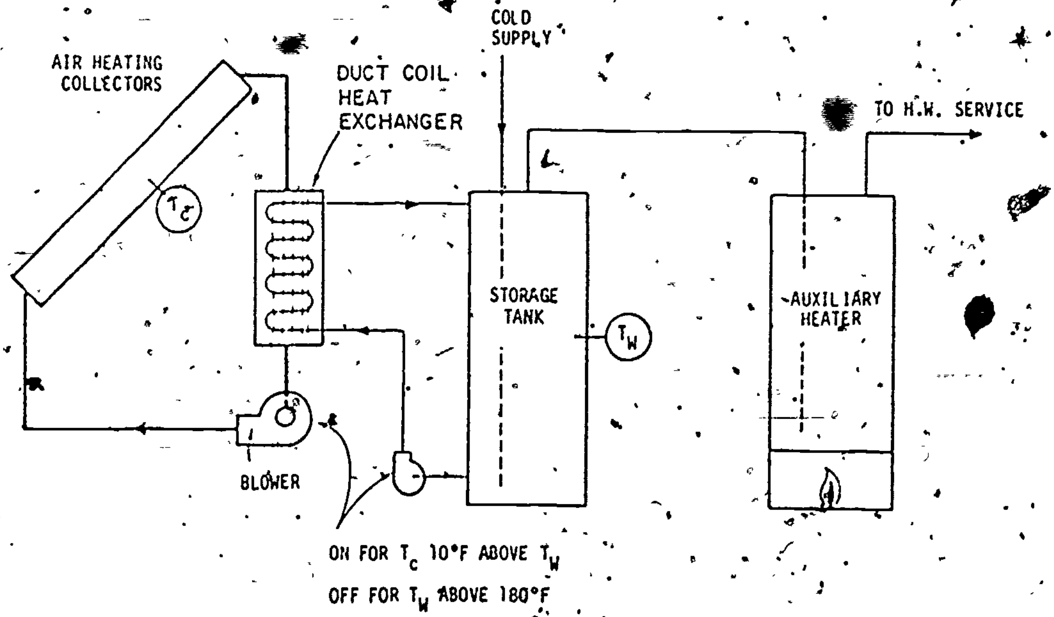
Figure 19-5: Dual Liquid Solar Hot Water Heater

the collector loop into the domestic water, even if pressure on the water side of the exchanger drops below that on the antifreeze side. A conventional tube-and-shell exchanger would therefore not usually be acceptable. Similarly, a coil inside the storage tank, through which the collector fluid is circulated, would not be satisfactory. Parallel tubes with metal bonds between them, so that perforation of one tube could not result in liquid entry into the other tube, would be a suitable design. A finned tube air-to-liquid heat exchanger could also be used by circulating the two liquids through alternate rows of tubes, heat transfer being by conduction through the fins.

Although aqueous solutions of ethylene glycol and propylene glycol appear to be most practical for solar energy collection, organic liquids such as Dowtherm J and Therminol 55 may be employed. Price and viscosity are drawbacks, but chemical stability and assurance against boiling are advantages over the antifreeze mixtures.

Solar Collection in Heated Air

In a manner similar to that described immediately above, solar energy can be employed in an air heating collector with subsequent transfer to domestic water in an air-to-water heat exchanger. Figure 19-6 illustrates a method for employing this concept. A solar air heater is supplied with air from a blower, the air is heated by passage through the collector, and the hot air is then cooled in the heat exchanger through which domestic water from a storage tank is either being pumped or is circulating by thermosiphon action. Air from the heat exchanger is recirculated to the collector. Differential temperature control (between collector and storage) is employed as in the other systems described. Advantages of the air


AIR HEATING COLLECTORS
T_C
DUCT COIL HEAT EXCHANGER
COLD SUPPLY
TO H.W. SERVICE
STORAGE TANK
T_W
AUXILIARY HEATER
BLOWER

ON FOR T_C 10°F ABOVE T_W

OFF FOR T_W ABOVE 180°F

Figure 19-6. Solar Hot Water Heater with Air Collectors

heat transfer medium are the absence of corrosion in the collector loop, freedom from liquid leakage, and freedom from boiling and loss of collector fluid. Disadvantages are the larger conduit between collector and heat exchanger, higher power consumption for circulation, and slightly larger collector surface requirements.

NON-CIRCULATING TYPE

Although probably of little potential interest in the United States, a type of solar water heater extensively used in Japan involves heat collection and water storage in the same unit. The most common type comprises a set of black plastic tubes about six inches in diameter and several feet long in a glass-covered box. Usually mounted in a tilted position, the tubes are filled each morning with water in which solar heat is collected throughout the day. The filling can be accomplished by a float-controlled valve and a small supply tank. Late in the day, heated water can be drained from the tubes for household use. In typical Japanese installations, non-pressurized hot water service is thus provided. Heat loss from the system is sufficiently high at night that hot water is usually not available until several hours after sunrise.

AUXILIARY HEAT

A dependable supply of hot water requires the availability of auxiliary heat for supplementing the solar source. The numerous methods of providing auxiliary heat vary in cost and effectiveness. A general principle for maximizing solar supply and minimizing auxiliary use is the avoidance of direct or indirect auxiliary heat input to the fluid entering the solar collector. If auxiliary heat is added to the solar hot water storage tank, so that the temperature of the liquid supplied to the collector is increased above that which only the solar system would provide, efficiency is reduced because of higher heat losses from the collector. Thus, auxiliary heat should be added at a point beyond (downstream from) the solar collector-storage system. Figures 19-3 and 19-4 show a conventional gas-fired hot water heater being supplied with hot water from the solar tank (whenever a hot water tap is opened). Any deficiency in temperature is made up by fuel in the thermostatted conventional heater. Alternatively, a "fast response", in-line heater can be employed. It is evident that auxiliary heat supply in these designs cannot adversely affect the operation of the solar system.

Another way in which auxiliary heat can be used without reducing solar collection efficiency is by electric resistance heaters in the upper portion of the solar storage tank, as shown in Figure 19-2. Temperature stratification in the tank, accomplished by bringing cold water from the main into the bottom and by circulating through the collector from the bottom of the tank to the upper portion of the tank, thereby prevents auxiliary heat from increasing the temperature of the water supplied to the collector. Water returning from the collector may be brought into the tank well below the level of the resistance heater (as shown by the dashed line), so that the hot supply is always available at the thermostatted temperature. In effect, the two tanks shown in Figures 19-3 and 19-4 are combined into one, with temperature stratification providing a separation. The total amount of storage is, of course, reduced unless the one tank is increased in size. If relatively high temperature water is desired, there may be an undesirable influence of auxiliary supply on collector efficiency because of some mixing in the tank.

Although the description of the above systems refers to direct circulation of water through the collector, the same factors apply to the systems involving heat exchange with antifreeze solutions or air circulating through the collector. In all cases, auxiliary heat should be supplied downstream from the solar storage tank, regardless of whether the water itself is circulated through the collector or whether heat is exchanged between the domestic water and a solar heat transfer fluid.

LOCATION OF COLLECTORS

If the slope and orientation of a roof is suitable, the most economical location for a solar collector in a residential water heating system is on the south-facing portion of the roof. The cost of a structure to support the collector is thereby eliminated, and pipe or duct connections to the conventional hot water system are usually convenient. In new dwellings, most installations can be expected on the house roof. Even in retrofitting existing dwellings with solar water heaters, a suitable roof location can usually be provided.

If the mounting of collectors on the roof is impractical, for any of several reasons, a separate structure adjacent to the house may be used. A sloping platform supported on a suitable foundation can be the base for the collector. Pumps, storage tank, and heat exchanger, if used, can be located inside the dwelling. Effective insulation on ducts and piping must be provided, however, so that cold weather operation will not be handicapped by excessive heat losses. In cold climates, collectors in which water is directly heated must be located so that drainage of the collector and exterior piping can be dependably and effectively accomplished.

TEMPERATURE STRATIFICATION IN SOLAR HOT WATER TANK

As in a conventional hot water heater, the temperature in the upper part of a solar hot water tank will normally be considerably higher than at the bottom. The lower density of hot water permits this stratification, provided that turbulence at inlet and outlet connections is not excessive. The supply of relatively cold water from the bottom of the tank to the collector permits the collector to operate at its highest possible efficiency under the prevailing ambient conditions. With a circulation rate such that a temperature rise through the collector of 15°F to 20°F occurs, the lower part of the storage tank is furnished to the collector for maximum effectiveness. If not much hot water is withdrawn from the tank during a sunny day, the late afternoon temperature at the bottom of an 80 gallon tank connected to a 40-to-50-square-foot collector may be well above 100°F -- even approaching the temperature in the top of the tank. Collection efficiency thus varies throughout the day, depending not only on solar availability but also on the temperature of water supplied to the collector from the tank bottom.

TEMPERATURE CONTROL LIMIT

In addition to the differential temperature control desirable in most solar water heating systems (which sense temperature difference between collector and storage), protection against excessive water temperature may be necessary. Several possible methods can be used. In nearly all types of systems, whether direct heating of the potable water or indirect heating through a heat exchanger, a thermostatically controlled mixing valve can be used to provide constant temperature water for household use.

Figure 19-7 illustrates one method by which this type of temperature control can be accomplished. Cold water is admitted to the hot water line immediately downstream from the auxiliary heater in sufficient proportion to secure the desired preset temperature. The solar hot water tank is allowed to reach any temperature attainable, and the auxiliary heater furnishes additional energy only when the auxiliary tank temperature drops below the thermostat set point. Maximum solar heat delivery is thus achieved, and no solar heat needs to be discarded except that which might sometimes be delivered when the main storage (preheat) tank is at the boiling point. Any additional solar heat collected under that condition would be dumped through a pressure relief valve, steam escaping to the surroundings. Figure 19-5 shows an optional second mixing valve for control of delivery temperature by admitting regulated amounts of solar heated water into the flow from the auxiliary heater.

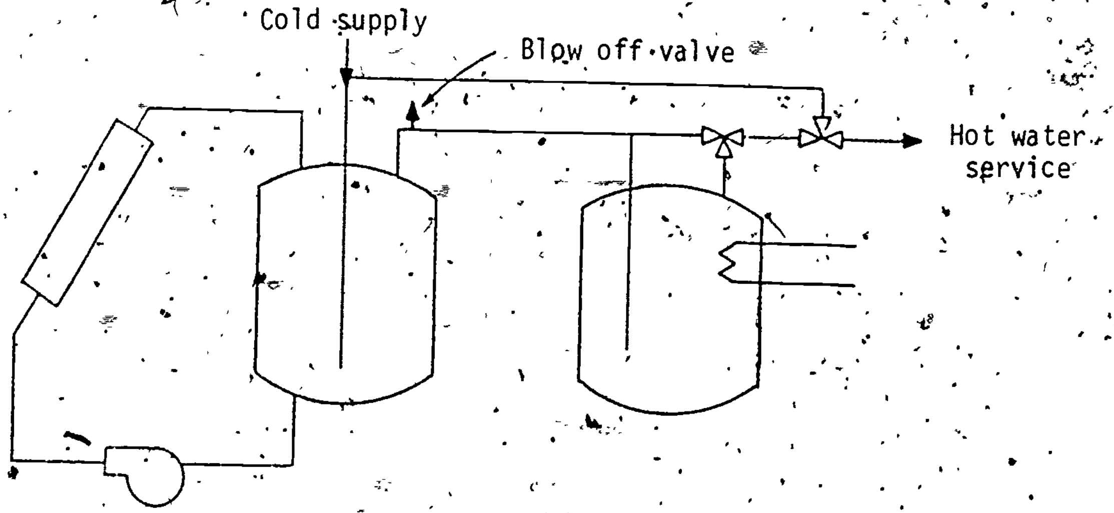


Figure 19-7. Direct Solar Water Heating with Mixing Valve

A steam vent from the solar hot water system involving a dual liquid design, with heat exchange, should normally be in the hot water loop rather than the collector loop. Loss of collector fluid by vaporization is thereby avoided. It is necessary, however, in this design, that the collector tubes and associated piping be capable of withstanding pressure at least as high as developed when the steam vent valve in the storage loop is actuated. If, for example, the blow-off valve in the storage circuit is set for 50 psi, and if the collector loop containing 50 percent ethylene glycol normally operates at a temperature 20°F above the storage tank temperature, pressure in the collector loop would also be about 50 psi when the storage tank vent is actuated. (Approximate equality of pressure is due to similarity between boiling point elevation and temperature difference in the heat exchanger.)

An alternative to the high pressure collector capability described above is available in the form of an organic heat transfer fluid having a high boiling point. Downtherm J or Therminol 55 have boiling points above 300°F, so if one of these fluids is used, the development of pressure in the collector loop would not occur, even when the storage system is venting steam at 50 psi. This option appears considerably more practical than the pressurized collector required with aqueous systems if the dual-liquid design is utilized.

Still another option for high-temperature protection is available if the collector is used as a heater for a high-boiling organic liquid or for air. To prevent the storage tank from reaching a temperature higher than desired, a limiting thermostat in that tank can be used simply to discontinue circulation of the heat transfer fluid (organic liquid or air) through the collector and heat exchanger. No additional heat is, therefore, dissipated in the form of collector heat loss. The collector temperature rises substantially, frequently above 300°F, but if properly designed, the collector suffers no damage. This system is probably the safest and most dependable of those herein described. With a reliable limit switch in the storage tank, there can be no dangerous pressure developments anywhere in the system. In addition, there is no loss of water (in the form of steam) even when there is no use of hot water for long periods.

If the hot water/cold water mixing valve downstream from the auxiliary heater is not used, a temperature limit control in the solar storage tank can be set at the maximum desired temperature of service hot water. Water, therefore, cannot be delivered at any temperature higher than the set point in the solar storage tank or the set point in the auxiliary heater, whichever is higher. Less solar storage capability would be involved in this design, however, because the solar storage tank is prevented from achieving higher temperatures, even when solar energy is available.

In a direct type of solar water heater operating at service pressure, with potable water circulating through the collector, a venting valve is provided near the top of the collector. It would have to be set for release at a pressure several pounds higher than the maximum in the service supply, so the collector storage system must withstand pressure usually above 50 psi. Occasional water loss through venting of steam would be expected.

If a non-pressurized direct type of solar water heater is used, with a float valve in the storage tank, the pressure relief valve can be set to operate at a pressure only slightly above atmospheric. Alternatively, the collector or storage tank may be continuously vented. Oversupply or under-use of solar heated water results in boiling and venting of the storage tank.

PERFORMANCE OF TYPICAL SYSTEMS

GENERAL REQUIREMENTS

A typical family of four persons requires, in the United States, about 80 gallons of hot water per day. At a customary supply temperature of about 140°F, the amount of heat required if the cold inlet is at 60°F is about 50,000 Btu per day.

There is a wide variation in the solar availability from region to region and from season to season in a particular location. There are also the short-term radiation fluctuations due to cloudiness and the day-night cycle.

Seasonal variations in solar availability result in a 200 to 400 percent difference in the solar heat supply to a hot water system. In the winter, for example, an average recovery of 40 percent of 1200 Btu of solar energy per square foot of sloping surface would require approximately 100 square feet of collector for the 50,000 Btu average daily requirement. Such a design would provide essentially all of the hot water needs on an average winter day, but would fall short on days of less than average sunshine. By contrast, a 50-percent recovery of an average summer radiant supply of 2000 Btu per square foot would involve the need for only 50 square feet of collector for satisfying the average hot water requirements.

It is evident that if a 50-square-foot collector were installed, it could supply the major part, perhaps nearly all, of the summer hot water requirements, but it could supply less than half the winter needs. If, on the other hand, a 100-square-foot collector were employed in order that winter needs could be more nearly met, the system would be oversized for summer operation and excess solar heat would have to be wasted. In such circumstances, if an aqueous collection medium were used, boiling of the system would occur and collector or storage venting of steam would have to be provided.

The more important disadvantage of the oversized collector (for summer operation) is the economic penalty associated with investment in a collector which is not fully utilized. Although the cost of the 100-square-foot collector would be approximately double that of the 50-square-foot unit, its annual useful heat delivery would be considerably less than double. It would, of course, deliver about twice as much heat in the winter season, when nearly all of it could be used, but in the other seasons, particularly in summer, heat overflow would occur. The net effect of these factors is a lower economic return, per unit of investment, by the larger system. Stated another way, more Btu per dollar of investment (hence cheaper solar heat) can be delivered by the smaller system.

As a conclusion to the above example, practical design of solar water heaters should be based on desired hot water output in the sunniest months rather than at some other time of year. If based on average daily radiation in the sunniest months, the unit will be slightly oversized and a small amount of heat will be wasted on days of maximum solar input. And quite naturally, on partly cloudy days during the season, some auxiliary heat must be provided. In the month of lowest average solar energy delivery, typically one-half to one-third as much solar heated water can be supplied, or actually the same quantity of water but with a temperature increase above inlet only one-half to one-third as high. Thus, fuel requirements for increasing the temperature of solar heated water to the desired (thermostatted) level could involve one-half to two-thirds of the total energy needed for hot water heating in a mid-winter month.

QUANTITATIVE PERFORMANCE

Although hundreds of thousands of solar water heaters have been used in the United States and abroad, quantitative performance data are extremely limited. In households where no auxiliary heat was used, the solar system probably supplied hot water most of the time, but failed during bad weather. If booster heat was used, hot water was always available, but the relative contributions of solar and auxiliary were seldom measured.

In a few research laboratories, particularly in Australia, some analytical studies of solar water heater performance, confirmed in part by experimental measurements, have been performed. More recently, analytical studies at the University of Wisconsin have been carried out. Table 19-1, based on an Australian study, shows the performance of a double-glazed, 45-square-foot solar water heater in several regions of the country. Variable solar energy and ambient temperature throughout the year result in 1.4 to 2.5 times as much solar heat supply to water in summer than in winter. Climatic differences produced a solar heat percentage ranging from 60 percent to 81 percent of the annual total hot water requirements. Table 19-2 shows monthly performance of the same system, in Melbourne, Australia, with average collection efficiency varying between 29 and 40 percent of incident radiation. Variation in inlet, outlet, and ambient temperature in a typical thermosiphon type of solar water heater is shown in Figure 19-8.

In a simulation study at the University of Wisconsin, hot water usage was programmed for a hypothetical residential user. The results show only slight variation in solar heat utilization at several use schedules and indicate only minor influence of storage temperature stratification on collector efficiency.

Table 19-1

Daily Means for Twelve Consecutive Months of Operation of Solar Water Heaters at Various Localities

Location	Adelaide	Brisbane*	Canberra	Deniliquin	Geelong	Melbourne	Sydney
Hot water discharge ** (gallons, US)	54.2	54.6	51.4	50.9	50.4	54.6	53.9
Electrical energy consumed (kWh)	3.5	2.5	3.4	2.5	3.8	4.6	4.4
Cold water temperature (°C)	17.7	21.6	12.7	16.8	15.9	16.1	16.6
Hot water temperature (°C)	58.9	56.4	58.4	60.3	58.7	57.4	57.7
Energy required to heat water (kWh)	9.8	8.4	10.3	9.7	9.5	9.9	9.8
Heat loss from storage tank (kWh)	2.2	1.9	2.5	2.5	2.2	1.9	1.9
Total energy consumed (kWh)	12.0	10.3	12.8	12.2	11.7	11.8	11.7
Solar energy contributed (kWh)	8.5	7.8	9.4	9.7	7.9	7.2	7.3
Solar energy contributed (%)	71.0	76.0	73.0	81.0	67.0	61.0	62.0
Solar contribution best month (%)	99.0	94.0	98.0	100.0	92.0	95.0	70.0
Solar contribution worst month (%)	47.0	57.0	43.0	57.0	45.0	38.0	51.0
Ratio best to worst	2.1	1.6	2.3	1.8	2.0	2.5	1.4

* Hail screens suspended above the absorbers. No correction made for reduction of absorbing area.

** Water discharged at 6:00 a.m. daily.

Double-glazed, flat-black, 45-square-foot solar collector tilted toward equator at latitude angle plus 2.5 degrees. Storage 84 gallons (US). Thermosiphon circulation. Electric auxiliary heat.

Table 19-2

Solar Water Heater Performance in Melbourne, Australia

Month	Mean Insolation on Absorber	Mean Daily Supplementary Energy	Mean Daily Solar Energy Contribution		System Efficiency
	Btu/ft^2 day	kWh	Percent	kWh	Percent
January	1630	2.9	75	8.9	40
February	2220	0.5	95	9.5	32
March	1690	2.6	74	7.4	33
April	1240	5.2	52	5.6	34
May	1290	6.2	47	5.5	32
June	1220	7.7	39	4.9	30
July	1290	8.1	38	5.0	29
August	1530	6.1	50	6.1	30
September	1600	4.9	59	7.1	33
October	1860	3.9	67	7.9	32
November	1880	3.7	68	7.9	32
December	1790	3.5	72	9.0	38
Year	1610	4.6	61	7.2	35

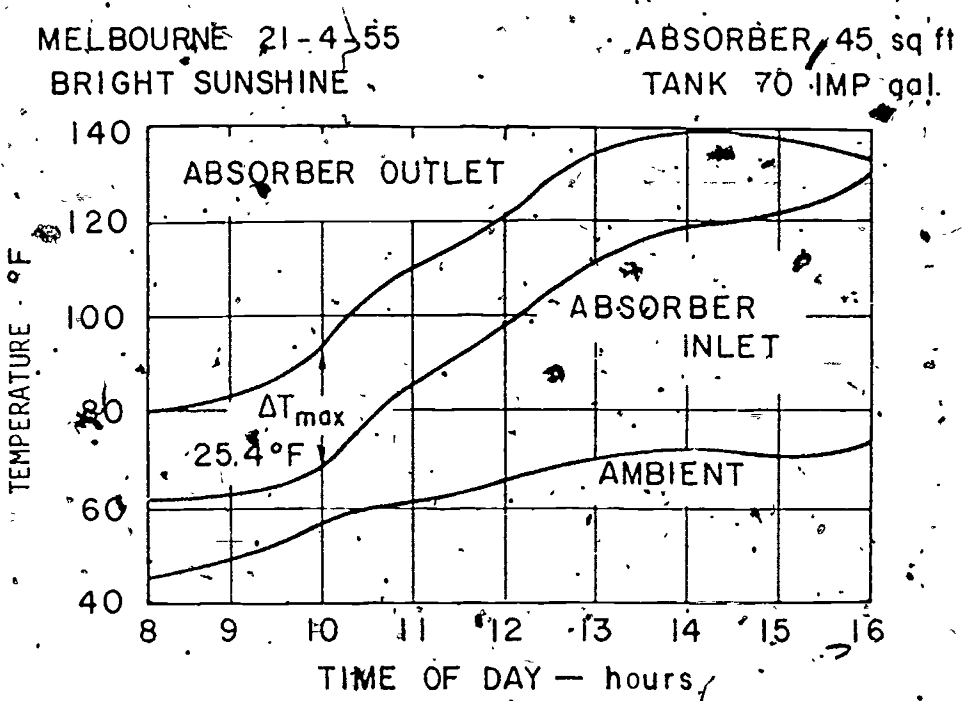


Figure 19-8. Absorber and Tank Temperatures for Thermosiphon Flow During a Typical Day

In summary, the normal output of well-designed solar water heating systems can be roughly estimated by assuming approximately 40 percent solar collection efficiency. Average monthly solar radiation multiplied by collector area and 40 percent delivery efficiency can provide a rough measure of daily or monthly Btu delivery. The total Btu requirements for the hot water supply, based on the volume used and the temperature increase set, then serve the basis for computation of percentage contribution from solar and the portion required to be supplied by fuel or electricity.

Sizing the Collectors

The curves shown in Figure 19-9 may be used to estimate the solar collector size required for hot water service in residential buildings having typical hot water systems. The system is assumed to be pumped liquid type, with liquid-to-liquid heat exchange, delivering hot water to scheduled residential uses from 6:00 a.m. until midnight. The shaded band represents results of computer calculations for eleven different locations in the United States. The cities included in the study are Boulder, Colorado; Albuquerque, New Mexico; Madison, Wisconsin; Boston, Massachusetts; Oak Ridge, Tennessee; Albany, New York; Manhattan, Kansas; Gainesville, Florida; Santa Maria, California; St. Cloud, Minnesota; and Washington, D.C. The hot water loads used in the computations range from 50 gallons per day (gpd) to 2000 gpd. The sizing curves are approximate and should not be expected to yield results closer than 10 percent of actual value.

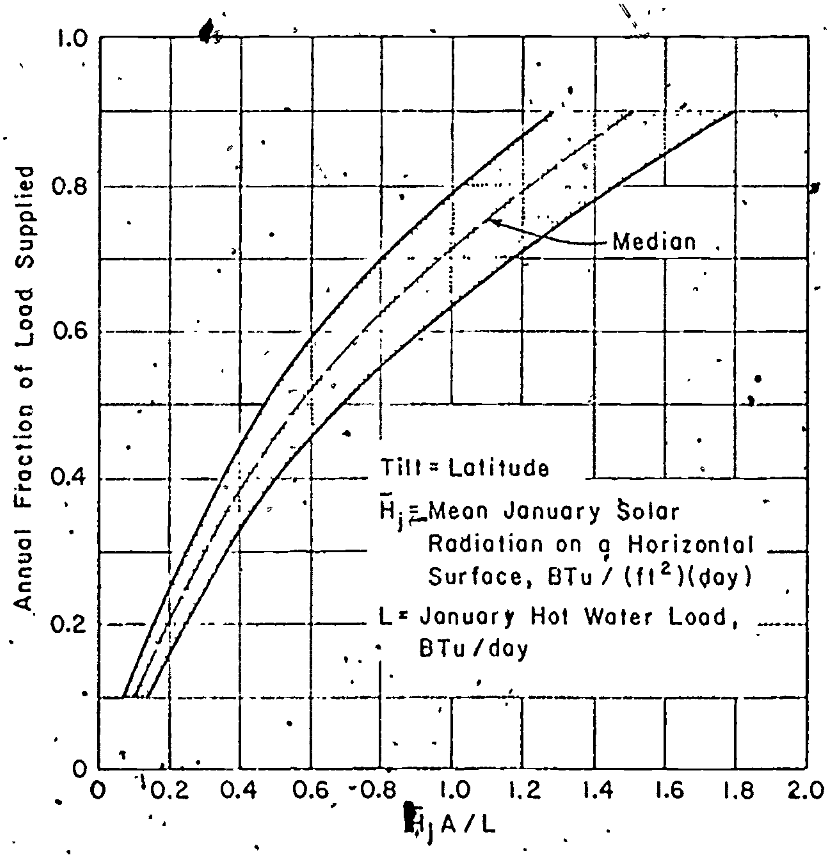


Figure 19-9. Fraction of Annual Load Supplied by Solar as a Function of January Conditions for Hot Water Heaters

The vertical axis shows the fraction of the annual water heating load supplied by solar. The horizontal axis shows values of the parameters, $\overline{H}_j A/L$, which involves the average daily January radiation on a horizontal surface, $\overline{H}_j$; the required collector area, A, to supply a certain percentage of the daily hot water load, L. The January average daily total radiation at locations in the United States can be estimated from the radiation map in Figure 19-10. Values on the map are given in Btu/(ft^2)(day). The curves are not applicable for values of f greater than 0.9.

It should be remembered that the service hot water load will be nearly constant throughout the year while the solar energy collected will vary from season to season. A system sized for January, with collectors tilted at the latitude angle, will deliver high temperature water and may even cause boiling in the summer. On the other hand, a system sized to meet the load in July will not provide all of the load in the winter months. Orientation of the collector can partially overcome month-to-month fluctuations in radiation and temperature.

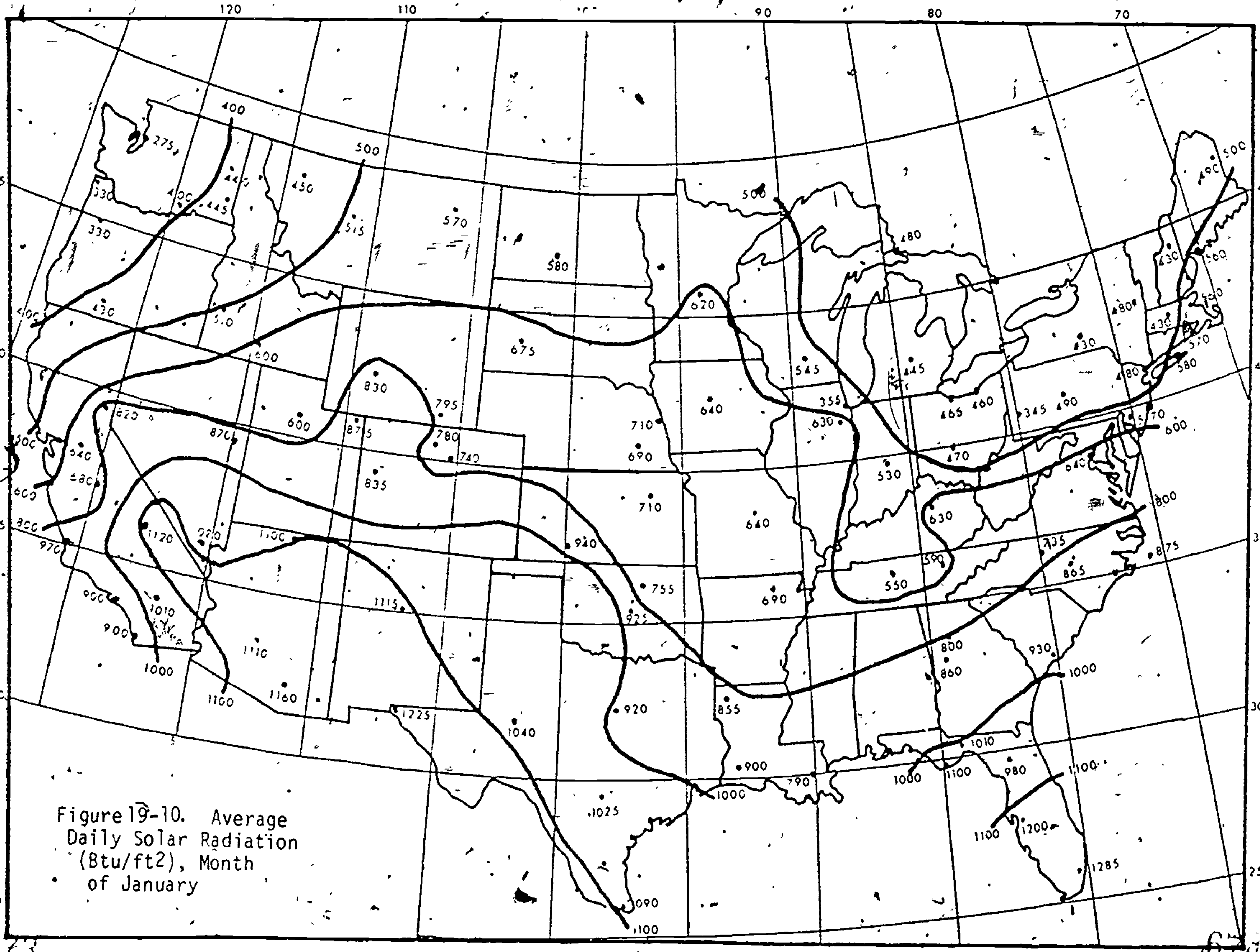
Sizing Examples

Example 19-1. Determine the approximate size of collector needed to provide hot water for a family of four in a residential building in Kansas City, Missouri.

Solution: The average daily service hot water load in January is:

L = 80 gallons/day x 8.34 pounds/gallon x 1 Btu/(lb)(°F) x (140°F - 50°F) = 60,048 Btu/day

The desired service water temperature is 140°F and the temperature of the cold water from the main is 50°F. The total average solar radiation, $\overline{H}_j$,


120
110
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970
1120
920
1100
1115
1110
1160
1100
1010
1000
900
900
1225
1040
920
855
900
1000
790
1025
1090
1100
800
860
930
1000
1010
1000
1100
980
1100
1200
1100
1285

Figure 19-10. Average Daily Solar Radiation (Btu/ft2), Month of January

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available in January, from Figure 19-10, is 680 Btu per square foot per day. For a water system to provide 60 percent of the annual load, from Figure 19-9, $\overline{H}_j A/L$ is about 0.8. Therefore:

$$A = 0.8 \times L/\overline{H}_j = (0.8 \times 60048)/680 = 70.6 \text{ square feet.}$$

If 3-by-8-foot collector modules are available, 2.9 units would be required. Three collector units should therefore be used.

Example 19-2. Determine the size of collector needed to provide hot water for a family of four in Albuquerque, New Mexico.

Solution: The monthly load will be approximately the same as in Example 19-1:

$L = 60{,}048$ Btu

From Figure 19-10, $\overline{H}_j = 1115$ Btu/(ft^2)(day). For a system to provide 60 percent of the annual load, Figure 19-9 shows that $\overline{H}_j A/L$ is approximately 0.8. The collector area required is:

$$A = (0.8 \times 60048)/1115 = 41.8$$

Using 3-by-6-foot collector modules, 2.3 units would be required for this system, either two or three modules should be used. If two modules are used, the system would be expected to provide less than 60 percent of the annual load.

COSTS

The cost of installing a solar water heater (exclusive of the hardware) may range from about \$300 for a system with a roof-mounted collector to over \$1000 for a collector mounted on a stand adjacent to a house. In a recent procurement of several types of solar water heaters for ground mounting next to existing houses, an electric utility company spent \$1500 to \$2000 for each system, including hardware, and totally installed.

Non-freezing collectors of about 50 square feet, 80-gallon water tanks, pumps, fans, and controls were included.

A solar collector manufacturer has announced the availability of a solar water heater "package" having a retail price of $995. The package consists of a 40-square-foot drainable collector, an 80-gallon storage tank, pumps, and controls. Installation and hook-up to the conventional system are not included.

As designs are standardized and manufacturing volume increases, it may be anticipated that the total installed cost of an average-sized residential solar water heating system will be less than $1000. Assuming a collector area of about 50 square feet and a reasonably sunny climate, this unit should be able to deliver at least 250,000 Btu per square foot of collector per year, for a total of 12.5 million Btu annually. With an average daily requirement for 50,000 Btu of heat for hot water, the 18 million Btu annually required could be two-thirds solar. If electric heat at five cents per kilowatt-hour (about $14 per million Btu) is being replaced, an annual electric saving of about $175 is achieved. A $1000 solar water heater could thus pay for itself from electric savings in about six years. Or, if conventionally financed at 8-percent interest, an annual cost of interest plus principal of, say, 12 percent, or $120 per year, would be less than the electric savings by something over $50 per year. This favorable economic comparison for solar water heaters is applicable now in many parts of the country and should prevail very generally in the next few years.

TRAINING COURSE IN
THE PRACTICAL ASPECTS OF
DESIGN OF SOLAR HEATING AND COOLING SYSTEMS
FOR
RESIDENTIAL BUILDINGS

MODULE 20

DESIGN CASE STUDY

SOLAR ENERGY APPLICATIONS LABORATORY
COLORADO STATE UNIVERSITY
FORT COLLINS, COLORADO

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TRAINEE OBJECTIVE

The objective of this module is to bring together the design procedures that have been presented during the previous sessions and integrate them into a complete design. At the end of this module the trainee should be able to design both air and water systems.

THE PROBLEM

The problem to be considered is the design of a solar system to provide space heating and service hot water for a residence to be located near Boulder, Colorado. The house is characterized by $UA = 900$ Btu/hr·°F and the service hot water load is 80 gallons per day to be raised from 52 °F to 140 °F. A liquid system is to be considered. The collector parameters are $F_R\overline{\tau\alpha} = 0.66$, $F_RU_L = 0.77$ Btu/hr·ft²·°F.

DESIGN PROCEDURE

The design procedure was outlined in Module 4. However, we wish to consider an additional step in the design process. That is, we want to determine the optimal mix of solar and energy conservation measures. Assume the following conditions: the cost for auxiliary fuel is $12 per million Btu and it is expected to increase at 10 percent per year; mortgage is for 20 years at an interest rate of 9 percent; the collector area-dependent system costs are to be $16/ft², and the system fixed costs are $5000. Finally, assume that we can reduce the building UA in increments of 100 Btu/hr·°F according to the following cost schedule:

UA Reduction Btu/hr·°F	Cost in $
900 to 800	$1000
800 to 700	1500
700 to 600	2000
600 to 500	2500
500 to 400	3000

The f-chart program was used to determine the economical investment that should be made for energy conservation in combination with the solar system. The objective was to minimize the present worth of the yearly total costs with the solar system and energy-conservation measures. The results from the first computation with no investment in energy conservation are shown in Table 20-1. From this table we see that the optimized collector area is 951 ft^2, the solar system will provide 77.3 percent of the annual load, and the present worth of the yearly total costs with and without solar are $39,446 and $58,438 respectively.

The effect of decreasing UA by 100 Btu/hr·°F may be determined by adding $1000 to the system fixed costs and repeating the analysis. This results in an optimized collector area of 855 ft^2, providing 77.4 percent of the annual load, and present worth values of $37,495 and $53,932 with and without the solar system respectively. We see from these results that it is economically advantageous to invest $1000 in energy-conservation measures to decrease the building UA from 900 to 800 Btu/hr·°F. Therefore, we should consider a further reduction in UA: with an additional cost of $1500 to achieve a further reduction by 100 Btu/hr·°F, the optimized area is 761 ft^2 which provides 77.6 percent of the annual load, and results in present worth factors of $36,135 and $49,927. From these results it is

Table 20-1. f-Chart Analysis

```
CODE        VARIABLE DESCRIPTION                                     VALUE UNITS

  1   AIR SYSTEM=1, LIQUID SYSTEM=2..........................      2.00
  2   COLLECTOR AREA.........................................    250.00 FT2
  3   FRPRIME-TAU-ALPHA PRODUCT(NORMAL INCIDENCE)............       .66
  4   FRPRIME-UL PRODUCT.....................................       .77 BTU/HR-F-FT2
  5   NUMBER OF TRANSPARENT COVERS...........................      2.00
  6   COLLECTOR SLOPE........................................     45.00 DEGREES
  7   AZIMUTH ANGLE (E.G. SOUTH=0, WEST=90)..................      0.00 DEGREES
  8   STORAGE CAPACITY.......................................     10.00 BTU/F-FT2
  9   EFFECTIVE BUILDING UA..................................    900.00 BTU/HR-F
 10   CONSTANT DAILY BLDG HEAT GENERATION....................      0.00 BTU/DAY
 11   HOT WATER USAGE........................................     80.00 GAL/DAY
 12   WATER SET TEMPERATURE..................................    140.00 F
 13   WATER MAIN TEMPERATURE.................................     52.00 F
 14   CITY CALL NUMBER.......................................     17.00
 15   PRINT OUT BY MONTH 1, BY YEAR=2........................      2.00
 16   ECONOMIC ANALYSIS YES 1, NO=2..........................      1.00
 17   USE OPTIMIZED COLLECTOR AREA 1, SPECIFIED AREA 2.......      1.00
 18   PERIOD OF THE ECONOMIC ANALYSIS........................     20.00 YEARS
 19   COLLECTOR AREA DEPENDENT SYSTEM COSTS..................     16.00 $/FT2 COLL
 20   CONSTANT SOLAR COSTS...................................   5000.00 $
 21   DOWN PAYMENT( OF ORIGINAL INVESTMENT)..................     90.00
 22   ANNUAL INTEREST RATE ON MORTGAGE.......................      9.00
 23   TERM OF MORTGAGE.......................................     20.00 YEARS
 24   ANNUAL NOMINAL(MARKET) DISCOUNT RATE...................      4.00
 25   EXPENSES(INSUR.,MAINT.) OF SYSTEM IN 1ST YEAR..........    100.00 $
 26   ANNUAL INCREASE IN ABOVE EXPENSES......................      6.00
 27   PRESENT COST OF AUXILIARY FUEL (CF)....................     12.00 $/MBTU
 28   CF RISE LINEAR 1, /YR 2,SEQ. OF VALUES 3...............      2.00
 29   IF 1, WHAT IS THE SLOPE OF CF INCREASE.................       .50 $/MBTU YR
 30   IF 2, WHAT IS THE ANNUAL RATE OF CF RISE...............     10.00
 31   ECONOMIC PRINT OUT BY YEAR 1, CUMULATIVE 2.............      2.00
 32   EFFECTIVE FEDERAL-STATE INCOME TAX RATE................     30.00
 33   TRUE PROP. TAX RATE PER $ OF ORIGINAL INVEST...........      0.00
 34   INCOME PRODUCING BUILDING YES 1 NO 2...................      2.00

BOULDER     CO 40.00

****ECONOMIC ANALYSIS****
OPTIMIZED COLLECTOR AREA      [illegible] FT2
INITIAL COST OF SOLAR SYSTEM($)    [illegible]
PRESENT WORTH OF YEARLY TOTAL COSTS WITH SOLAR($)         38446.
PRESENT WORTH OF YEARLY TOTAL COSTS W/O SOLAR($)          [illegible]

****THERMAL ANALYSIS****
 TIME PERCENT  INCIDENT HEATING WATER DEGREE  AMBIENT
      SOLAR    SOLAR    LOAD    LOAD  DAYS    TEMP
               (MBTU)   (MBTU) (MBTU) (F DAY) (F)
 YR    77.3    [illegible]  112.56  21.47  [illegible]
```

apparent that a further reduction in UA is in order. A reduced UA of 600 Btu/hr·^{o}F resulted in an optimized area of 668 ft^2, providing 78 percent of the annual load, and present worth factors of $35,365 and $46,421. Because the present worth with the solar system is still decreasing, additional reduction in UA of 100 Btu/hr·^{o}F, for an additional cost of $2500, was considered. The analysis resulted in an optimized collector area of 586 ft^2, providing 79 percent of the annual load, and present worth factors of $35,192 and $4[illegible]inally, a further reduction in UA of 100 Btu/hr·^{o}F at an increase in cost of $3000 resulted in an optimized collector area of 515 ft^2, providing 80.9 percent of the annual load and present worth factors of $35,638 and $40,911.

Although the present worth of yearly total costs without solar is still decreasing, the system with solar is still less expensive than the system without solar. Therefore it is clear that the building should include energy-conservation measures at a total cost of $7000 and consider a solar system with optimal collector area of 586 ft^2. The results of the analysis are summarized in Table 20-2.

Assuming the collector flow rate to be 0.02 gal/min per square foot of collector, the flow rate through the collectors should be about 12 gallons per minute. The pressure drop through the collectors, lines, fittings, and heat exchanger must now be calculated in order to size the pump. These calculations may be performed by reference to Figures 16-1 and 16-2 (for the system schematic), to Figures 16-4 and 16-5 (to calculate pressure drops in lines and fittings), to Figures 16-6 through 16-9 (to determine pressure drop in the collector-storage heat exchanger), and to Figure 16-10 (or equivalent, to select the pump).

Table 20-2. Summary of Economic Analysis

UA Btu/hr·ft^2	Investment in Energy Conservation Measures, $	Solar Collector Area ft^2	Percent of Annual Load Supplied by Solar	Present Worth With Solar $	Present Worth Without Solar $
900	0	951	77.3	39,446	58,438
800	1000	855	77.4	37,495	53,932
700	2500	761	77.6	36,135	49,927
600	4500	668	78.0	35,365	46,421
500	7000	586	79.0	35,192	43,416
400	10000	515	80.9	35,638	40,911

AIR SYSTEM

Suppose that an air system had been desired rather than a liquid system. If the parameters which describe collector performance are the same as for the liquid system, the results would remain the same. That is, we would want 586 ft^2 of collector. Consequently, the storage should contain approximately 293 ft^3 of 3/4" to 1" washed rocks. The rocks should not be more than 35 percent fractured, 80 percent 3/4", with a minimum amount of fines. The rocks must be clean and free of dirt. The storage container should be constructed according to the diagram shown in Figure 20-1. The depth of the rocks should be 5.5 ft, the length should be 8.9 ft, and the width should be 6 ft.

The system schematic is shown in Figure 20-2. The main blower should deliver approximately 1172 cfm.

The ductwork, as indicated in Figure 20-2, should be sized for low static pressure drop to minimize electrical power requirements. The recommended sizing requirements are that the flow velocity in the ducts should not exceed 600-800 feet per minute. Figure 16-11 may be used to determine the duct size required to meet these specifications. The ductwork should be insulated with at least 1 inch of 2-lb. insulation, and turning vanes should be used in all elbows. All joints must be carefully sealed to prevent air leaks. The system is shown schematically in Figure 20-3. The service hot water system is shown in Figure 20-4.

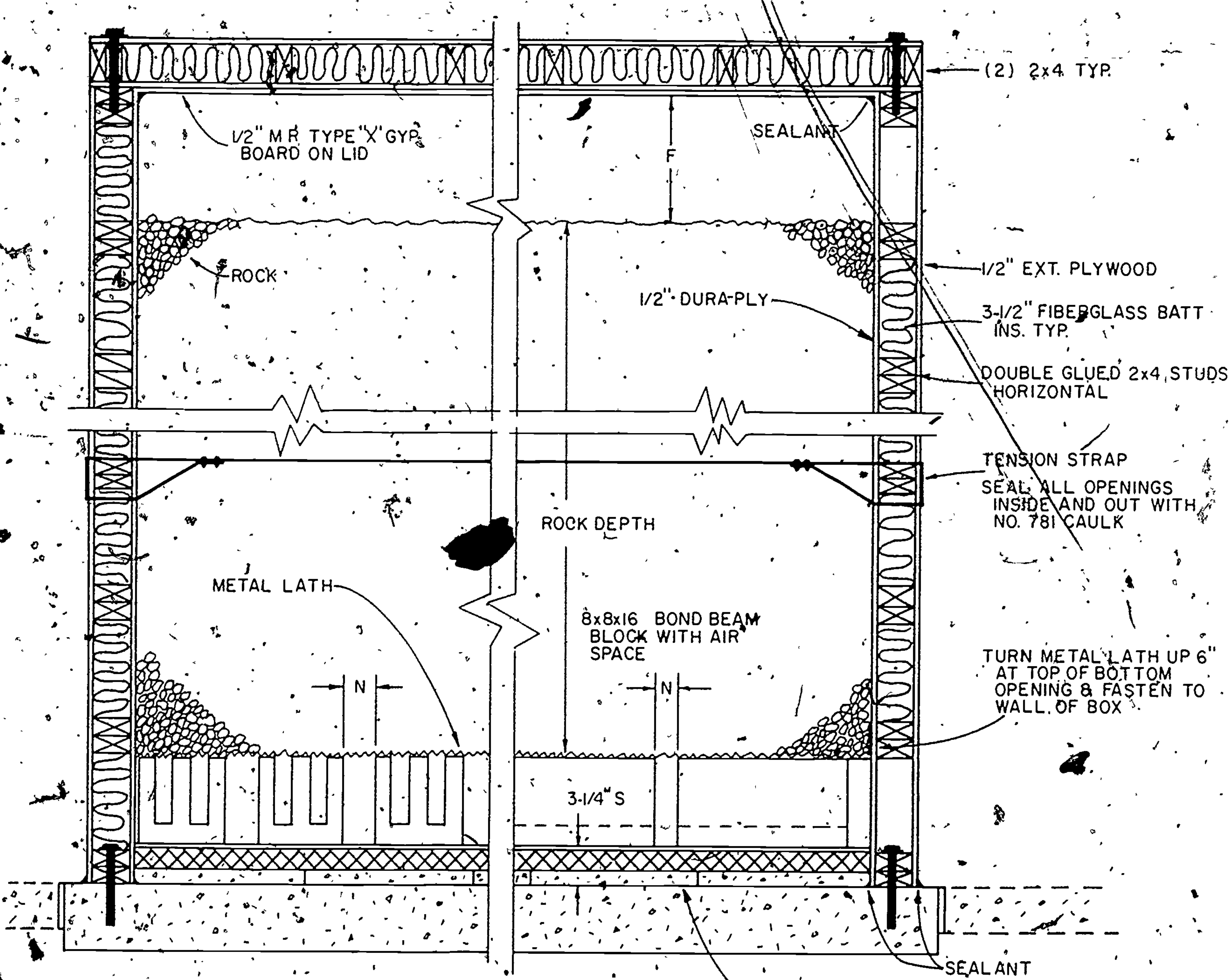
(2) 2x4 TYP.
1/2" M R TYPE 'X' GYP. BOARD ON LID
SEALANT
F
ROCK
1/2" EXT. PLYWOOD
1/2" DURA-PLY
3-1/2" FIBERGLASS BATT INS. TYP.
DOUBLE GLUED 2x4 STUDS HORIZONTAL
TENSION STRAP
SEAL ALL OPENINGS INSIDE AND OUT WITH NO. 781 CAULK
ROCK DEPTH
METAL LATH
8x8x16 BOND BEAM BLOCK WITH AIR SPACE
N
N
TURN METAL LATH UP 6" AT TOP OF BOTTOM OPENING & FASTEN TO WALL OF BOX
3-1/4" S
SEALANT
1-1/2" CAP TYPE CONC. BLOCK SPACER

Figure 20-1. Rock-Bed Storage

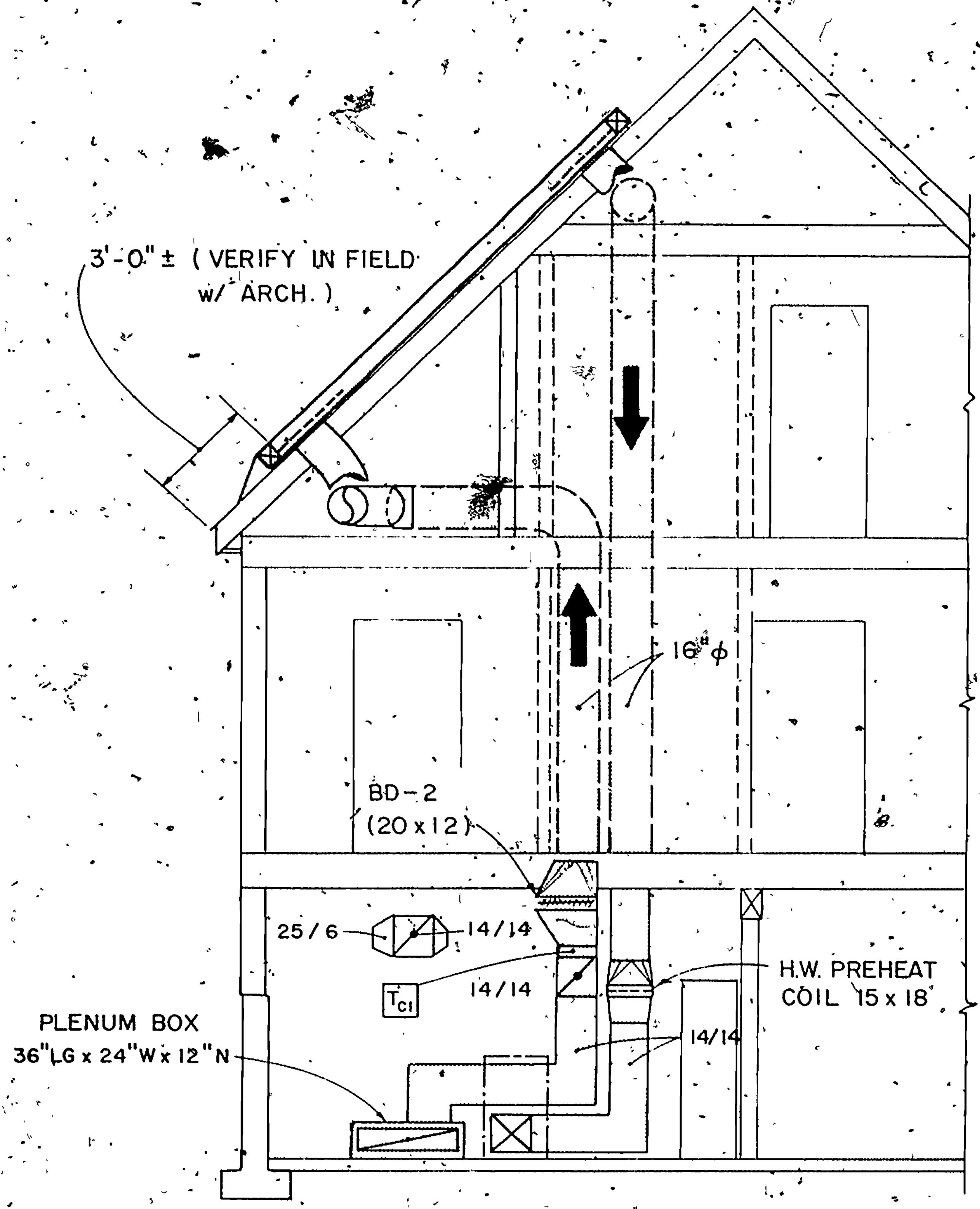
3'-0" ± (VERIFY IN FIELD w/ ARCH.)
16" φ
BD-2 (20 x 12)
25 / 6
14 / 14
T_{CI}
14 / 14
H.W. PREHEAT COIL 15 x 18
14/14
PLENUM BOX 36" LG x 24" W x 12" N

Figure 20-2. Air System Schematic

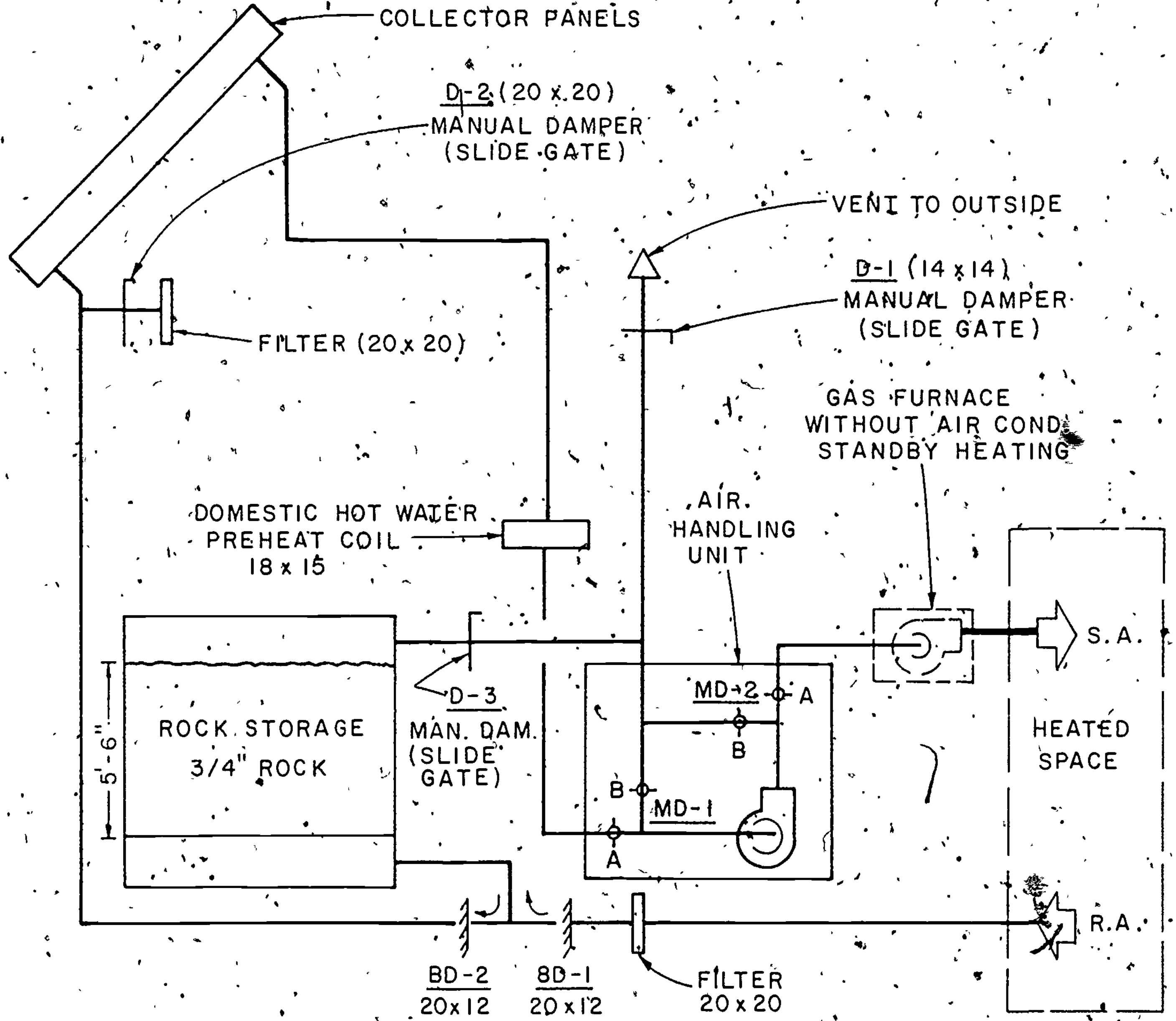
COLLECTOR PANELS
D-2 (20 x 20)
MANUAL DAMPER
(SLIDE GATE)
VENT TO OUTSIDE
D-1 (14 x 14)
MANUAL DAMPER
(SLIDE GATE)
FILTER (20 x 20)
GAS FURNACE
WITHOUT AIR COND.
STANDBY HEATING
AIR
HANDLING
UNIT
DOMESTIC HOT WATER
PREHEAT COIL
18 x 15
S.A.
ROCK STORAGE
3/4" ROCK
5'-6"
D-3
MAN. DAM.
(SLIDE
GATE)
MD-2
A
B
HEATED
SPACE
B
MD-1
A
R.A.
BD-2
20x12
BD-1
20x12
FILTER
20 x 20

Figure 20-3. Air System With Air Handler

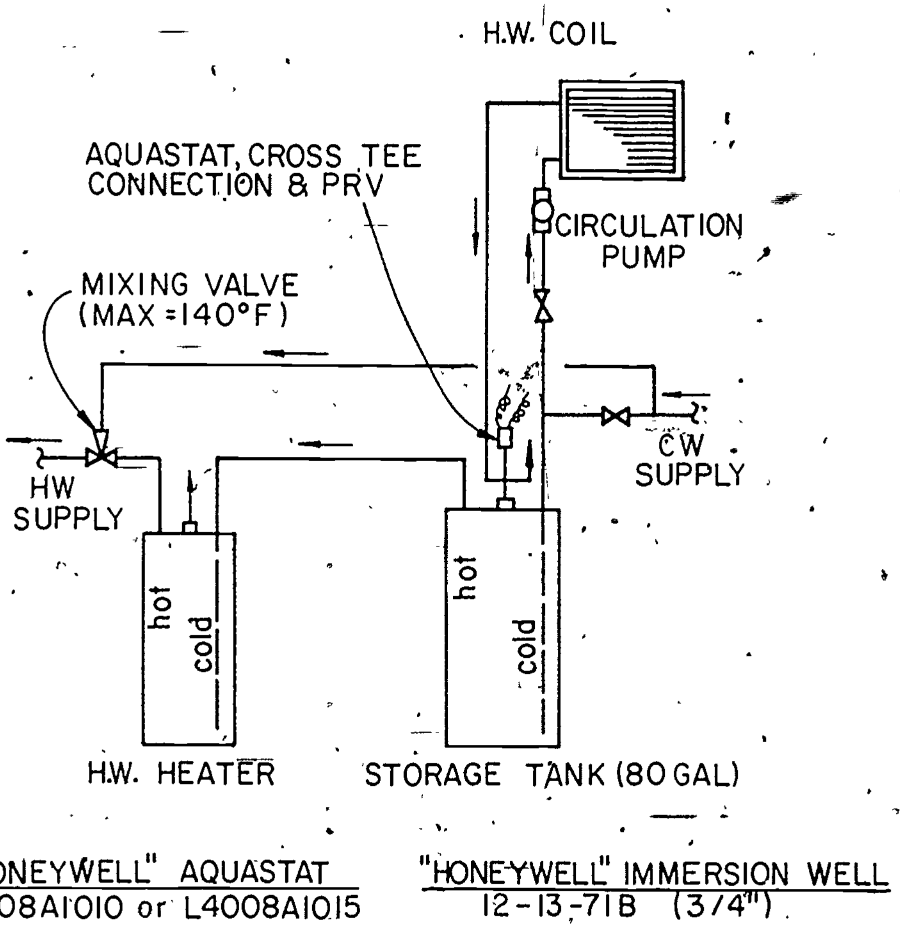
H.W. COIL
AQUASTAT, CROSS TEE CONNECTION & PRV
CIRCULATION PUMP
MIXING VALVE (MAX = 140°F)
HW SUPPLY
CW SUPPLY
hot
cold
hot
cold
H.W. HEATER
STORAGE TANK (80 GAL)
"HONEYWELL" AQUASTAT
L6008A1010 or L4008A1015
"HONEYWELL" IMMERSION WELL
12-13-71B (3/4")

Figure 20-4. Service Hot Water System

TRAINING COURSE IN
THE PRACTICAL ASPECTS OF
DESIGN OF SOLAR HEATING AND COOLING SYSTEMS
FOR
RESIDENTIAL BUILDINGS

MODULE 21

STRUCTURAL, MECHANICAL AND SCHEDULING CONSIDERATIONS.

SOLAR ENERGY APPLICATIONS LABORATORY
COLORADO STATE UNIVERSITY
FORT COLLINS, COLORADO

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INTRODUCTION

Structural and mechanical considerations for home construction as well as for the solar system are discussed in this module along with the scheduling of sequential and concurrent activities for installing solar heating and/or cooling systems in new home construction. Undoubtedly, standard or simplified critical path schedules, which are used in this module, are used by contractors only for construction of large building projects. Nevertheless, two example schedules for constructing typical homes are presented as illustrations, and items related to the solar systems are discussed with respect to structural and mechanical considerations. If attention is not given to the sequence of assembly, strength of supports, and details of connections, unnecessarily difficult situations could result when systems are installed.

OBJECTIVE

The objective in this module is to familiarize trainees with important items in construction and sequences that should be followed for installation of solar systems in new homes.

CONSTRUCTION SCHEDULE FOR A TYPICAL HOME WITH AN AIR-HEATING SOLAR SYSTEM

PART 1, ROCK-BED STORAGE

The initial steps in the construction of a home with an air-heating solar system is shown in part 1 of a construction schedule which is Figure 21-1. The building contains a basement in this example, and the principal solar system component included in this phase of construction is a rock-bed storage unit located in the basement. The construction activities concerning the pebble-bed storage unit are identified by heavy lines in the figure.

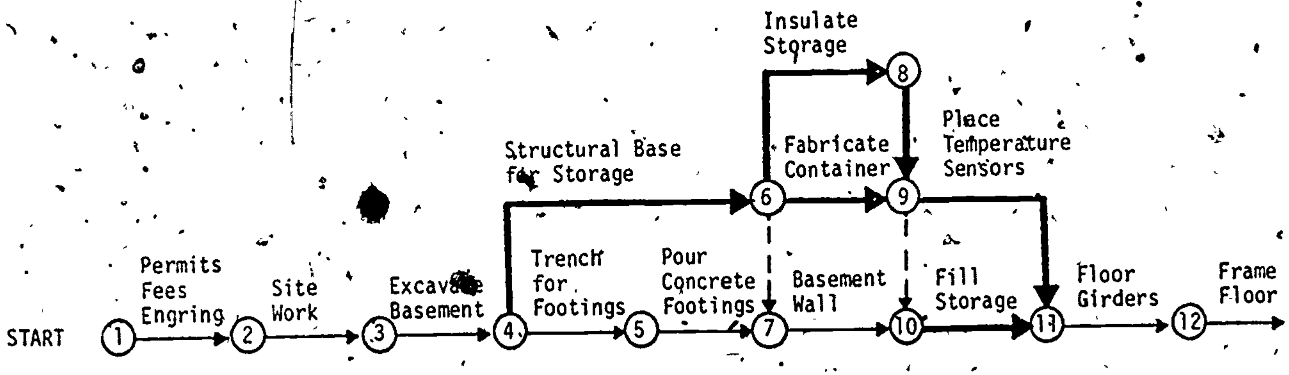
Insulate Storage
Structural Base for Storage
Fabricate Container
Place Temperature Sensors
START
Permits Fees Engring
Site Work
Excavate Basement
Trench for Footings
Pour Concrete Footings
Basement Wall
Fill Storage
Floor Girders
Frame Floor
1
2
3
4
5
6
7
8
9
10
11
12

Figure 21-1. Part 1 - Pebble-Bed Storage Fabrication.

The structural base for the rock-bed storage unit should be constructed during the foundation work of the building. The concrete base on which the storage bin is to be constructed should be scheduled for pouring along with the concrete footings. If the storage container walls are to be concrete, the rock bed can be located in the corner of the basement to utilize common walls. If the container is to be fabricated of wood, the walls and insulation can be constructed prior to placement of the floor girders and joists.

The recommended rock depth in a storage bin is 5 to 6 feet. The weight of rocks is approximately 100 lbs/ft^3, and so the weight of rocks on the foundation slab will be 500 to 600 pounds per square foot, and the weight of the container and the bond beams in the rock bin must be added. Assuming an additional 100 lbs loading on the foundation slab, the total load could be as much as 700 lbs/ft^2. A concrete slab at least 8 inches in thickness, reinforced with wire mesh, is recommended to support the load.

The ducts connected to the top and bottom plenums of the rock bin should be a diffuser, particularly to aid the flow of air from the rock bin into the ducts. Because air flows in both the top and bottom ducts are bi-directional; a diffuser is needed on both. Without diffusers, some portion of the rock bin, particularly if the box is rectangular, may not be effective for passage of air, hence for storing and reclaiming the heat.

In fabricating a rock box, it is important to consider the lateral stresses exerted by the box on the sides. Considering a natural angle of repose of one-inch gravel to be about 40^{0}, there could be as much as 1500 pounds of force on each linear foot of wall with a rock depth of 6 feet. To support these internal rock forces on the walls of the box, steel rock ties within the box at one-third depth and again at two-thirds depth should be used. Without the tie rods, the walls of the box could expand with the force, and the joints could crack, and air and heat losses could occur.

Rocks must not be dumped into the storage bin, because of the tie rods in the box, and also the rocks could fracture into smaller pieces and clog the interstices of the rock bed. A tightly packed rock bed leads to greater pressure drops to circulate the air through storage.

The placement of temperature sensors for the control system and, if desired, for monitoring purposes, is a simultaneous activity with the filling of the rock bin. It is not practical to install sensors after the gravel has been placed in the bin.

PART 2, COLLECTOR SUPPORTS

The support structure for solar collector modules or panels may be the vertical wall of the building or the roof trusses or rafters. The schedule in Figure 21-2 assumes that collectors are to be placed on the roof, but may be revised as necessary for attachment to the external wall. The spacing between roof trusses, or wall studs, should be convenient for the type of collector to be used in the solar system.

The weight of flat-plate solar collectors can vary considerably, but a typical collector might weigh 5 to 6 pounds per square foot. When collectors are placed on the roof, the added load is a consideration to the design of roof trusses. Although the pitch of most roofs will be steep, snow and ice loads should not be discounted in sizing the roof truss. In addition, the steeper pitched roof will be subjected to greater positive and negative loads due to wind forces. In areas where high winds are possible, structural designs of the roofs should consider these factors.

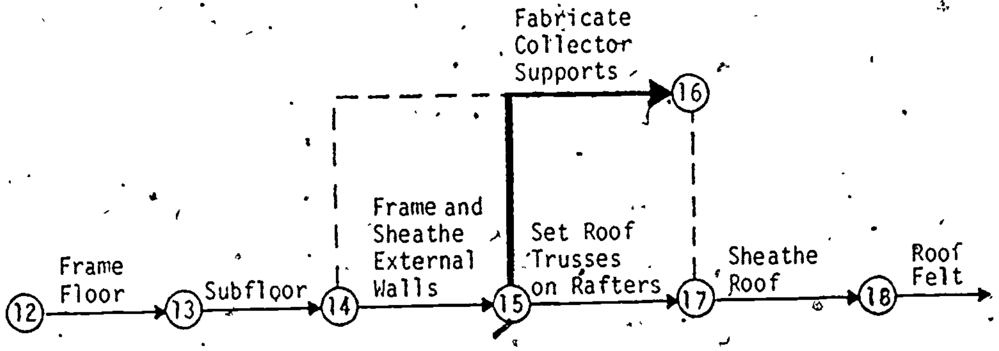
Fabricate Collector Supports
16
Frame and Sheathe External Walls
Set Roof Trusses on Rafters
Sheathe Roof
Roof Felt
12
Frame Floor
13
Subfloor
14
15
17
18

Figure 21-2. Part 2 - Collector Support Construction

Time can be saved in mounting collectors if forethought is given to convenient placement of purlins and nailers. There should be space provided for manifold air ducts in the attic which will cross the roof trusses. Because roof trusses are largely pre-assembled, they should be made up with cross-pieces that will support the manifolds.

The placement of manifolds is at the convenience of the designer. The guideline to follow is to minimize lengths of large sizes to minimize cost, but at the same time, greater pressure drops in long lengths of small ducts should be avoided.

PART 3, INSTALLATION OF COLLECTORS, PIPING, AND CONTROL PANEL

Installation of collector modules can be scheduled simultaneously with the roofing and flashing as shown in Figure 21-3. The collectors, in most instances, will replace the roofing, and should be rendered watertight with cap strips of the collector array.

For heavy collector modules, a mechanical hoist such as a fork lift may be needed for installation. Although detailed instructions may be provided by the manufacturer for assembly of collector modules, considerable attention should be given to effect air-tight joints at all duct connections. Air leakage into the collector array can cause excess heating of the rooms because the quantity of air leak into a system must also flow out of the system; and usually that occurs through the dampers that control the flow of heat into the rooms. Ultimately, the heat flows outdoors from the rooms.

The preheat tank and plumbing connections to the air-water heat exchanger can be scheduled with other plumbing in the building. The heat exchanger, which is in the duct connected to the collector manifold, will be installed with the duct work. All plumbing should be leak-tested after installation.

If the control panel for the solar system is a separate unit from, say the air handler, the installation can be scheduled with the other rough electrical work. The control panel should be located close to the solar system for convenience of wiring, system checks and maintenance purposes.

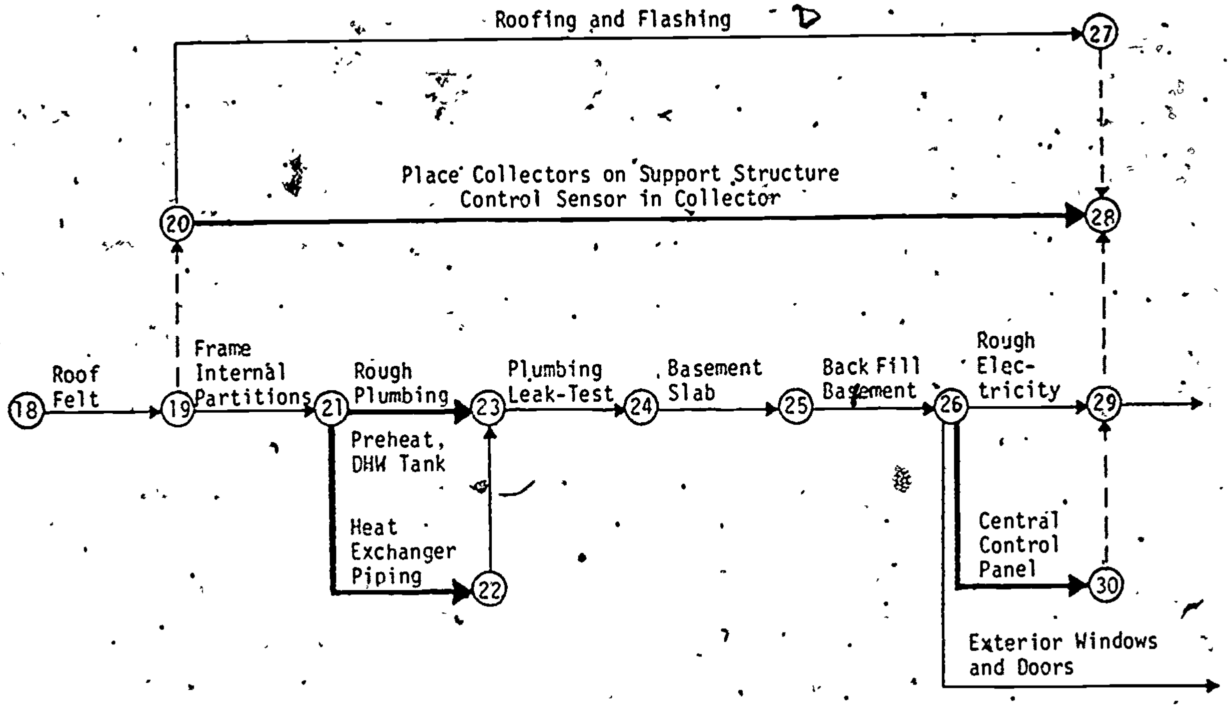
Roofing and Flashing
Place Collectors on Support Structure
Control Sensor in Collector
Roof Felt
Frame Internal Partitions
Rough Plumbing
Preheat, DHW Tank
Heat Exchanger Piping
Plumbing Leak-Test
Basement Slab
Back Fill Basement
Rough Elec-tricity
Central Control Panel
Exterior Windows and Doors
18
19
20
21
22
23
24
25
26
27
28
29
30

Figure 21-3. Part 3 - Collector, Heat Exchanger and Control Panel Installation

PART 4, INSULATION

Insulation on piping and ducts can be applied following leak-tests. Insulation should cover valves as well as the piping. Loosely wrapped insulation may allow air circulation and therefore is not effective, but tightly wrapped insulation reduces the thickness and is therefore poor practice. All ducts and pipes, whether they are flexible or rigid, should be insulated.

PART 5, CONNECTING THE CONTROLS

Connecting the control wires is virtually the last activity in the installation of the solar system before the system is checked out. Usually the company that provides the control unit will have full instructions for making the connections.

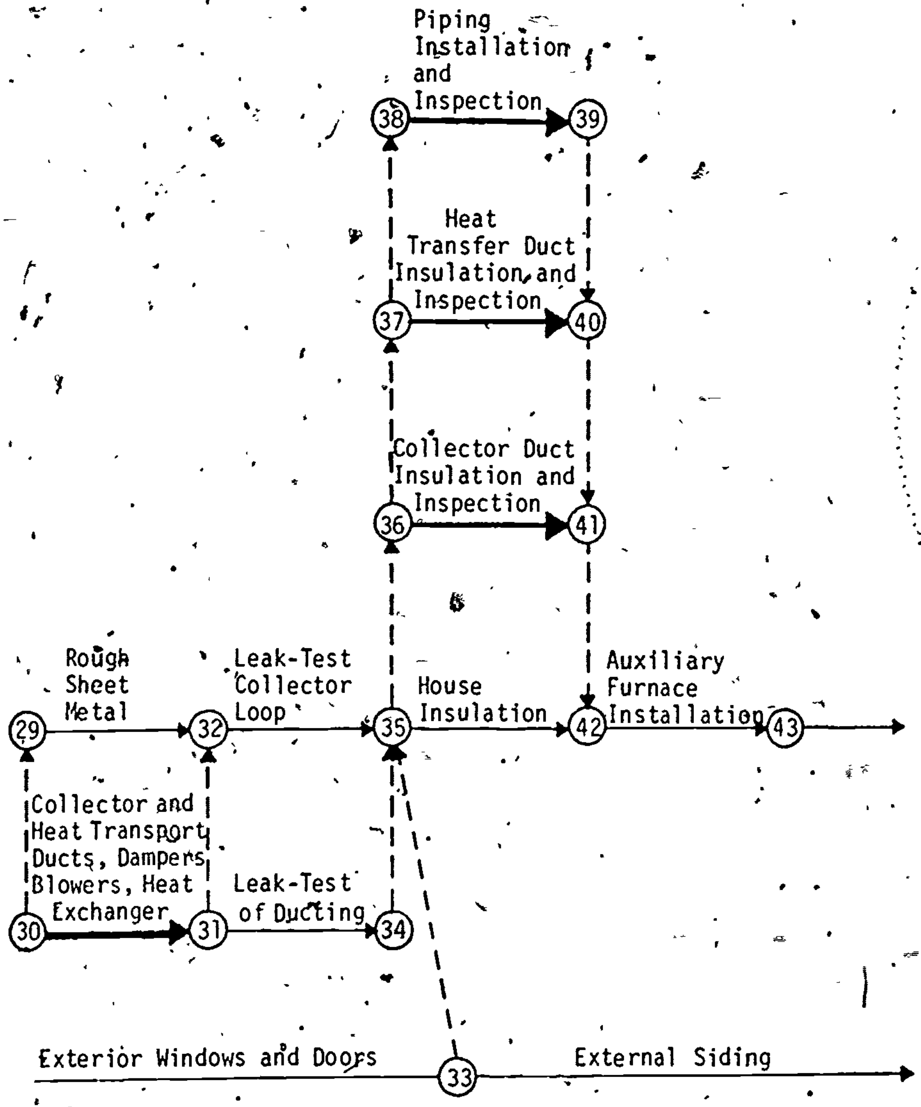
Piping Installation and Inspection
38
39
Heat Transfer Duct Insulation and Inspection
37
40
Collector Duct Insulation and Inspection
36
41
Rough Sheet Metal
29
32
Leak-Test Collector Loop
35
House Insulation
42
Auxiliary Furnace Installation
43
Collector and Heat Transport Ducts, Dampers, Blowers, Heat Exchanger
30
31
Leak-Test of Ducting
34
Exterior Windows and Doors
33
External Siding

Figure 21-4. Part 4 - Application of Insulation

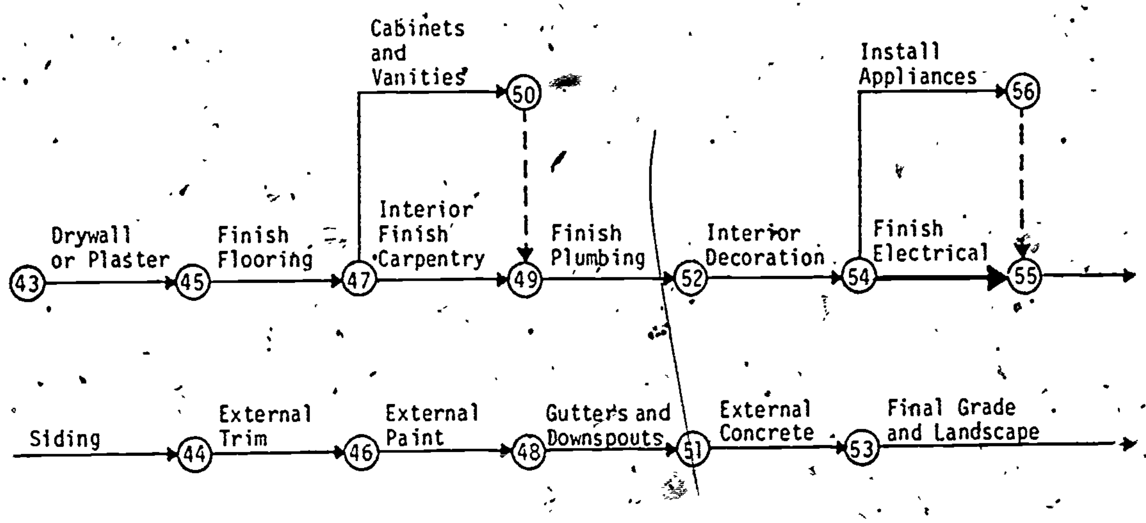
Cabinets and Vanities
Install Appliances
Drywall or Plaster
Finish Flooring
Interior Finish Carpentry
Finish Plumbing
Interior Decoration
Finish Electrical
43
45
47
49
50
52
54
55
56
Siding
External Trim
External Paint
Gutters and Downspouts
External Concrete
Final Grade and Landscape
44
46
48
51
53

Figure 21-5. Part 5 - Connecting Controls

PART 6, FINISH THE HEATING SYSTEM AND FINAL INSPECTION

After installation is completed, the system should be tested to be sure that all modes operate as desired, that is, the dampers are open or shut as they should be, and the blower is activated properly. If necessary, jumper cables can be used across terminals to check out the system. If dampers do not close firmly, there will be leaks into the flow loop, and when cold air is mixed with the warm air, considerable temperature degradation can take place. Although heat may not be lost from the system, lowered air temperatures can cause the auxiliary furnace to operate a larger portion of the time than is actually necessary. Connections to the arms of motorized dampers may become loose with time, causing dampers not to close properly. Frequent checks may be necessary to insure proper closure.

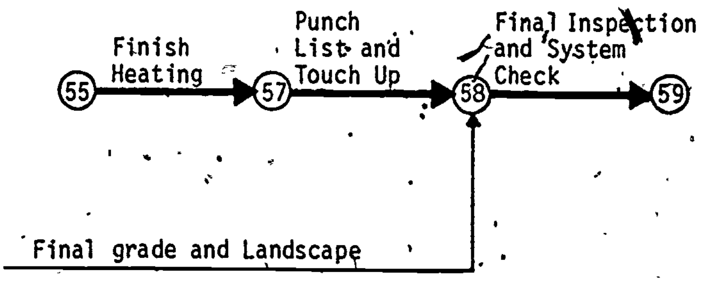
Finish Heating
Punch List and Touch Up
Final Inspection and System Check
55
57
58
59
Final grade and Landscape

Figure 21-6. Part 6 - Finishing and Final Inspection

CONSTRUCTION SCHEDULE FOR A TYPICAL HOME WITH A TYPICAL LIQUID-HEATING SOLAR SYSTEM

PART 1; WATER STORAGE TANK

The structural base for the thermal storage unit is provided when the concrete is poured for the footings. A thicker concrete slab than a normal basement floor should be prepared for the storage foundation.

A prefabricated tank is recommended for the storage vessel which should be placed on the base before the floor girders are assembled. The storage tank should be provided with appropriate connections for pipes and the control sensor. Depending upon the type of storage tank, the bottom insulation should be installed before placement to eliminate extra work later to insulate the tank.

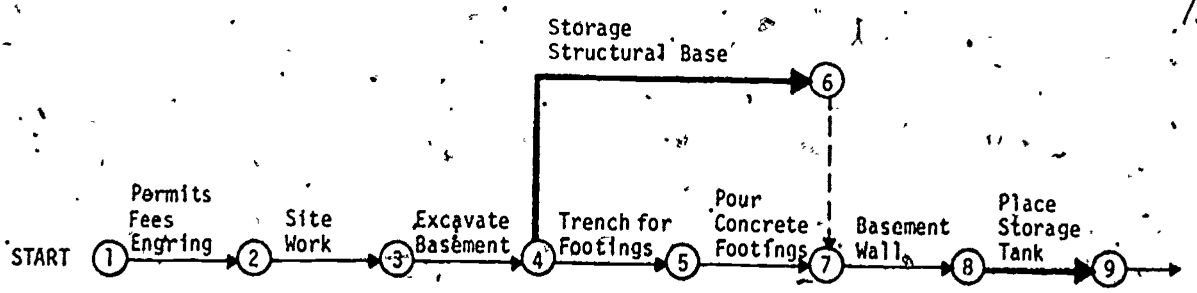
Storage Structural Base
6
START
1
Permits Fees Engrng
2
Site Work
3
Excavate Basement
4
Trench for Footings
5
Pour Concrete Footings
7
Basement Wall
8
Place Storage Tank
9

Figure 21-7. Part 1 - Storage Tank Foundation

PART 2, COLLECTOR SUPPORTS

Normally the rafters are the supports for the collector array; however, special supports may be required for some installations. When rafters are to support the collectors directly, some preplanning will reduce the labor costs to assemble and secure the collectors. Normally collectors are mounted on plywood sheathing and the collectors are secured by bolts through the plywood.

Pipe manifolds are normally placed along the top and bottom of the collector array. Provisions for easy access, not only for installation, but also for maintenance, should be provided because replacement of flexible connections between the collector outlets and the manifold is a common maintenance item and, although replacement is simple, it can be made difficult with restrictive access.

Collectors mounted on flat roofs will require supports to tilt the collectors at a desired angle. The supports should be secured to the rafters, and open collector supports should be closed in to prevent wind drag and snow drifting, both of which will add extra loads on the roof. Consideration should be given to place the bottom of collectors off the roof surface by 12 to 18 inches for installations in regions where there is heavy snowfall, because as the snow piles up at the base of collectors, the absorber plate can be shaded; unusual snowfall removal may be advisable.

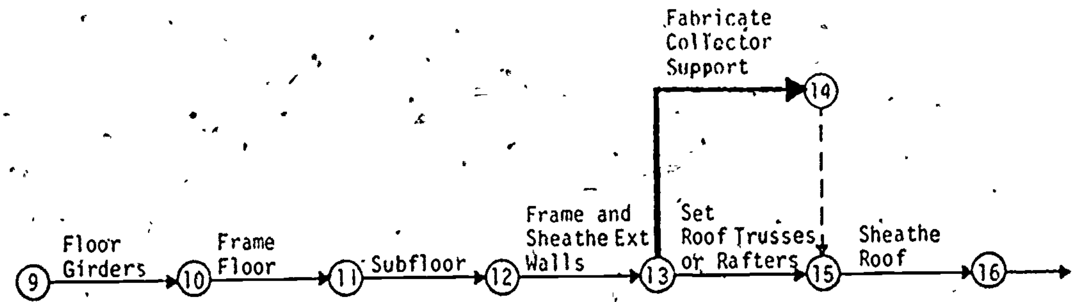
Fabricate Collector Support
14
Floor Girders
Frame Floor
Subfloor
Frame and Sheathe Ext Walls
Set Roof Trusses or Rafters
Sheathe Roof
9
10
11
12
13
15
16

Figure 21-8. Part 2 - Fabricate Collector Support

PART 3, COLLECTOR INSTALLATION AND PIPING

Collectors should be carefully inspected before installation. Broken glass, improper seals, absorber plate conditions, and bad plumbing fittings are easy to identify. There is an advantage in placing liquid collectors tightly together side-by-side to minimize side heat losses from each collector module. When this cannot be done, insulation between the collector modules should be used to reduce the side heat losses. After the headers are connected to the collectors, they should be leak-tested. Replacement of flexible connections, or tightening of joints, is easiest during collector installation and before cap strips are placed over collector joints.

Rough plumbing for the solar system is scheduled with the normal house plumbing, and the control sensors can be placed along with the rough plumbing. The filter unit, all the valves, the heat exchanger, pumps, and an expansion tank should be installed in the collector loop and the entire system leak-tested.

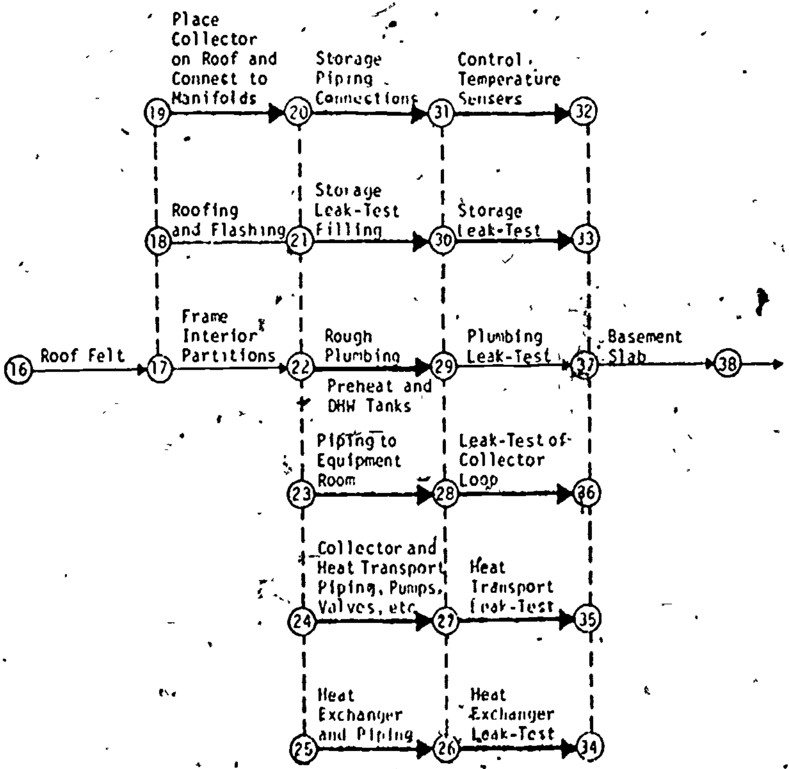
Place Collector on Roof and Connect to Manifolds
Storage Piping Connections
Control Temperature Sensors
Roofing and Flashing
Storage Leak-Test Filling
Storage Leak-Test
Roof Felt
Frame Interior Partitions
Rough Plumbing
Plumbing Leak-Test
Basement Slab
Preheat and DHW Tanks
Piping to Equipment Room
Leak-Test of Collector Loop
Collector and Heat Transport Piping, Pumps, Valves, etc.
Heat Transport Leak-Test
Heat Exchanger and Piping
Heat Exchanger Leak-Test
16
17
18
19
20
21
22
23
24
25
26
27
28
29
30
31
32
33
34
35
36
37
38

Figure 21-9. Part 3 - Collection Installation and Piping

PART 4, INSULATION AND AUXILIARY BOILER

The pipes in the solar system should be insulated to minimize heat losses, and the insulating must be done before drywalling. The storage tank, heat exchanger, and the expansion tank, as well as the valves, should be well insulated.

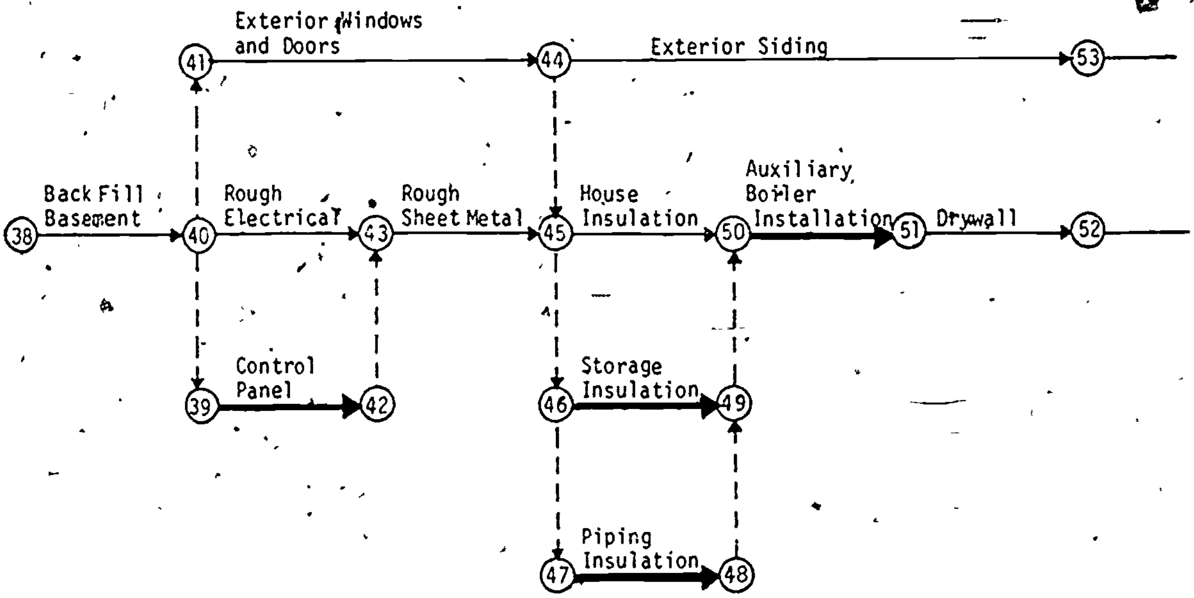
Exterior Windows and Doors
Exterior Siding
Back Fill Basement
Rough Electrical
Rough Sheet Metal
House Insulation
Auxiliary Boiler Installation
Drywall
Control Panel
Storage Insulation
Piping Insulation
38
39
40
41
42
43
44
45
46
47
48
49
50
51
52
53

Figure 21-10. Part 4 - Insulation and Auxiliary Boiler

PART 5, PREHEAT TANK AND CONTROL WIRING

The control panel—wiring is the final item related to installation of the solar system. It is recommended that initial tests be made of the solar system and final inspection and tests be made after a short period of system operation.

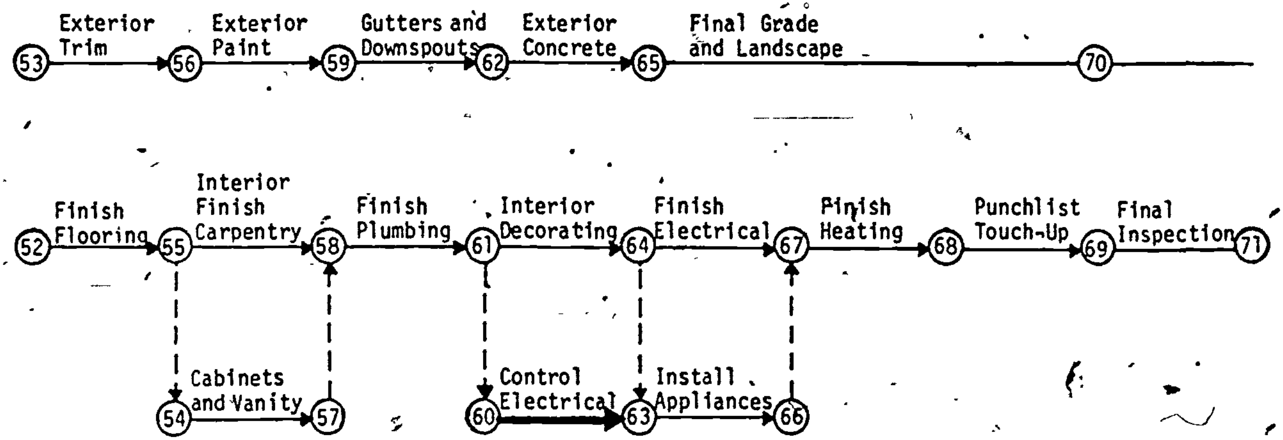
Exterior Trim
Exterior Paint
Gutters and Downspouts
Exterior Concrete
Final Grade and Landscape
53
56
59
62
65
70
Finish Flooring
Interior Finish Carpentry
Finish Plumbing
Interior Decorating
Finish Electrical
Finish Heating
Punchlist Touch-Up
Final Inspection
52
55
58
61
64
67
68
69
71
Cabinets and Vanity
Control Electrical
Install Appliances
54
57
60
63
66

Figure 21-11. Control Connections

TRAINING COURSE IN
THE PRACTICAL ASPECTS OF
DESIGN OF SOLAR HEATING AND COOLING SYSTEMS
FOR
RESIDENTIAL BUILDINGS

MODULE 22

FUTURE PROSPECTS FOR SOLAR HEATING AND COOLING SYSTEMS

SOLAR ENERGY APPLICATIONS LABORATORY
COLORADO STATE UNIVERSITY
FORT COLLINS, COLORADO

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INTRODUCTION

The solar systems that are described in other modules of this manual are cost-effective systems that have been installed and operated. Data obtained from experimental systems indicate that they function satisfactorily in residential buildings. Fluids that are heated by solar energy in flat-plate collectors are sufficiently high in temperature to heat space and hot water and to provide the hot water to drive an absorption cooling machine. Although efficiencies of the systems vary, they are generally about 30 percent and, while such an efficiency is satisfactory, if it can [illegible] improved by better components at lower energy cost, the improvements are worthwhile. A number of new features and components of systems are being researched and many could improve system performance significantly. Flat-plate collectors can be improved with selective coatings or redesigned to provide greater efficiencies in heat collection. Storage with latent heat materials could provide greater heat capacity in more compact space, and storage for liquid systems with direct contact heat exchanger to eliminate some hardware would improve system performance. If air conditioning equipment using solar-heated air could be developed, the air-heating solar systems could be used throughout the year for heating and cooling. These and many other future prospects are in store for solar heating and cooling systems.

OBJECTIVE

This module describes some prospective features and components in solar heating and cooling systems that could improve overall system performance. The objective of the trainee is to know some of the new features that could become economical to add to the systems described in this course and to recognize that considerable research and development effort is being devoted to component hardware in solar heating and cooling systems.

SOLAR COLLECTORS

The most important component in a solar system which could improve performance is the solar collector. Improvements which will increase efficiency of energy collection and reduce the delivered costs are practically worthwhile. Among many interesting possibilities are the addition of selective surfaces to absorbers, and collectors with the air evacuated from around the absorber plates to reduce heat losses and improve collector efficiency.

SELECTIVE SURFACES

Selective surfaces have high absorptance of solar radiation and low emittance of long-wave radiation. There are a variety of selective surfaces that could be used on flat-plate collectors, and some are being tested on experimental units. Several coatings such as copper oxide and black nickel have been available for a long time, but technical problems and cost have limited their use. Black chrome appears to hold some promise and some flat-plate collectors are presently available with such absorber coatings. Characteristics of some selective surfaces are listed in Table 22-1.

Table 22-1

Selective Surfaces Characteristics

Coating	Absorptance	Emittance
Converted Zinc	0.90	0.071
Black Nickel	0.88	0.066
Black Chrome	0.92	0.085

EVACUATED TUBE COLLECTORS

Evacuation of the air around the absorber plate is potentially a significant improvement in solar collectors. There are a number of different designs that are being assembled and tested, and at least one manufacturer makes them in moderate quantities. Evacuated collectors will produce more useful heat than standard flat-plate collectors under the same sun and weather conditions because the losses from the absorber are greatly reduced. With a vacuum surrounding the absorber, conduction and convection losses are effectively negligible and, if the absorber coating is a selective surface, the radiation loss is small.

One design, by Corning Glass Works, is shown in Figure 22-1. Inside an evacuated glass tube which is four inches in diameter is a copper absorber plate with a selective surface. Bonded to the plate is a copper U-tube which carries the heat transfer fluid. The ends of the tube protrude through one end of the glass tube, and the absorber plate is free to expand toward the other end. The efficiency range of the collector varies from about 75 percent when the inlet fluid temperature is low to about 60 percent when the fluid is near the boiling temperature of water. Most flat-plate collectors have high efficiency with low inlet fluid temperatures, but have low efficiencies when the fluid temperature is near 200°F. The evacuated tube collector has a

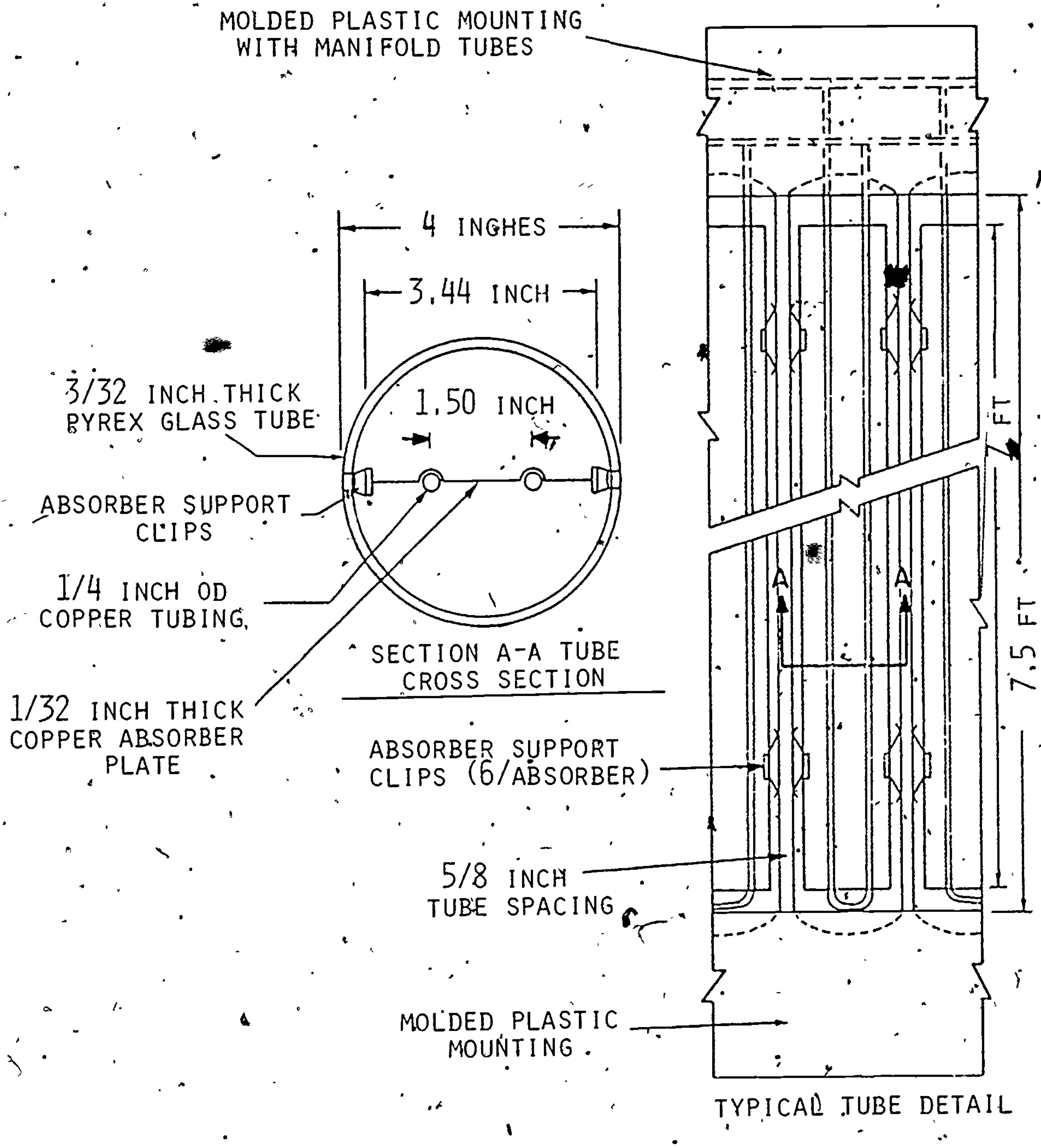
MOLDED PLASTIC MOUNTING WITH MANIFOLD TUBES
4 INCHES
3.44 INCH
3/32 INCH THICK PYREX GLASS TUBE
1.50 INCH
ABSORBER SUPPORT CLIPS
1/4 INCH OD COPPER TUBING
1/32 INCH THICK COPPER ABSORBER PLATE
SECTION A-A TUBE CROSS SECTION
A
A
FT
7.5 FT
ABSORBER SUPPORT CLIPS (6/ABSORBER)
5/8 INCH TUBE SPACING
MOLDED PLASTIC MOUNTING
TYPICAL TUBE DETAIL

Figure 22-1. Corning Glass Company Evacuated Tube Collector

significant advantage when producing high temperature heat to the system and can be used effectively with solar cooling units where high temperature fluid is needed.

An evacuated tube collector design by the Owens-Illinois Glass Company is shown in Figure 22-2. There are three concentric glass tubes with the middle one coated with a black selective surface. The vacuum is between the outer and middle tubes. Fluid is transported through the inner tube and, as it passes through the annulus in contact with the absorber tube, heat is transferred from the glass to the fluid..

Two other evacuated tube collectors are being experimentally tested, one by the General Electric Company for use in air-heating systems and another is by the Philips Company in West Germany for liquid systems. Many variations in design of evacuated tube collectors are possible; and different designs will gradually advance to the practical stage.

CONCENTRATING COLLECTORS

Concentrating collectors are used when very high temperature fluid is needed to drive heat engines or to be used in industrial processes. If concentrating collectors can be designed to be more efficient than flat-plate collectors, operate reliably, and with little maintenance so that the cost of delivering energy is low, then such collectors can have potential uses in residential solar systems. Experience thus far has indicated otherwise, but there is considerable research underway and new designs for concentrating collectors are being developed.

One type of low concentration collector is being developed by the Northrup Company and is being tested on a number of solar systems for large buildings. A linear focusing collector with a Fresnel lens is the type being developed and is shown in Figure 22-3. The collector is

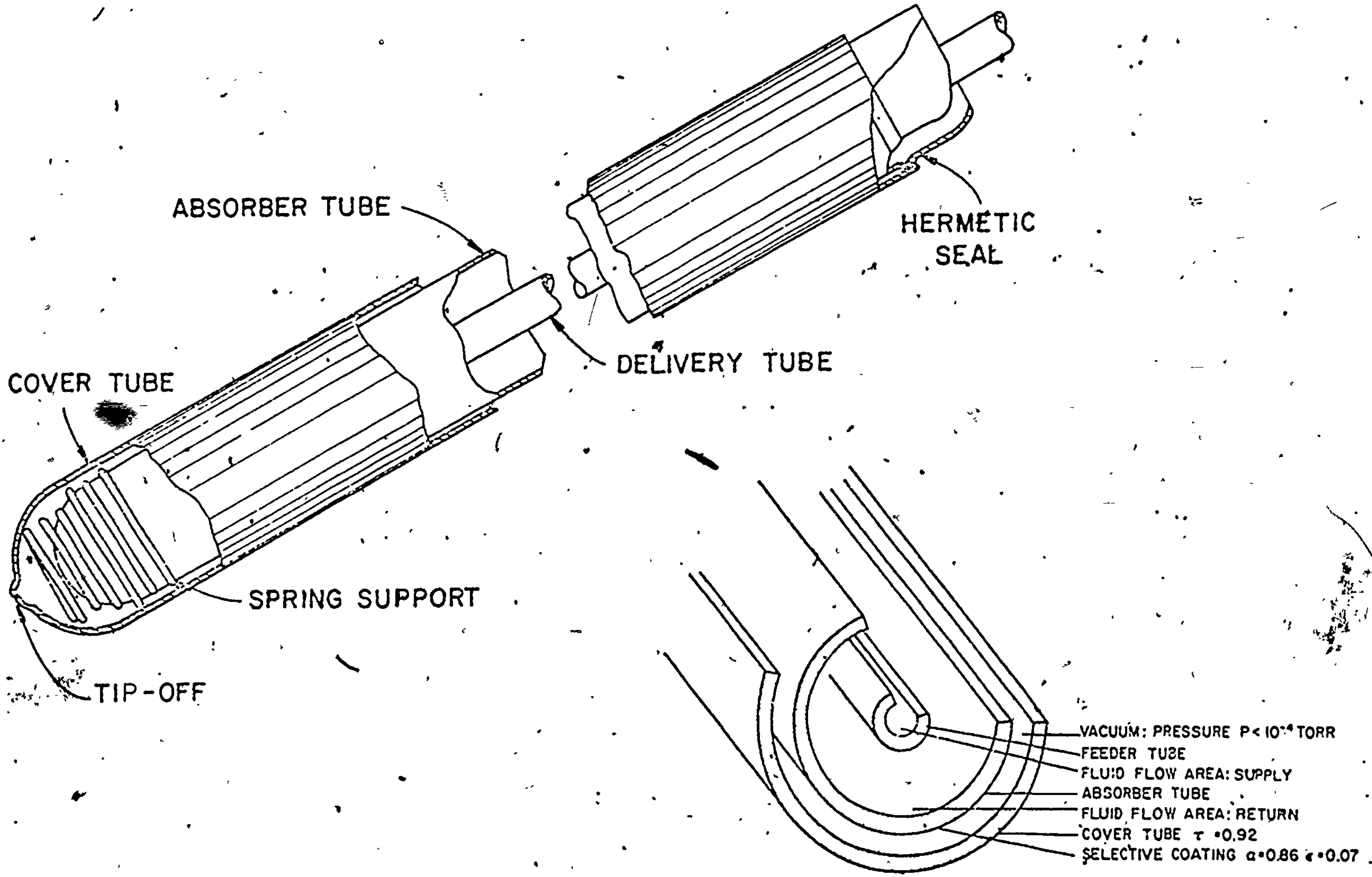
ABSORBER TUBE
HERMETIC SEAL
DELIVERY TUBE
COVER TUBE
SPRING SUPPORT
TIP-OFF
VACUUM: PRESSURE P< 10⁻⁴ TORR
FEEDER TUBE
FLUID FLOW AREA: SUPPLY
ABSORBER TUBE
FLUID FLOW AREA: RETURN
COVER TUBE τ =0.92
SELECTIVE COATING α=0.86 ε=0.07

Figure 22-2. Schematic of the Owens-Illinois [illegible]vacuated Tube Solar Collector

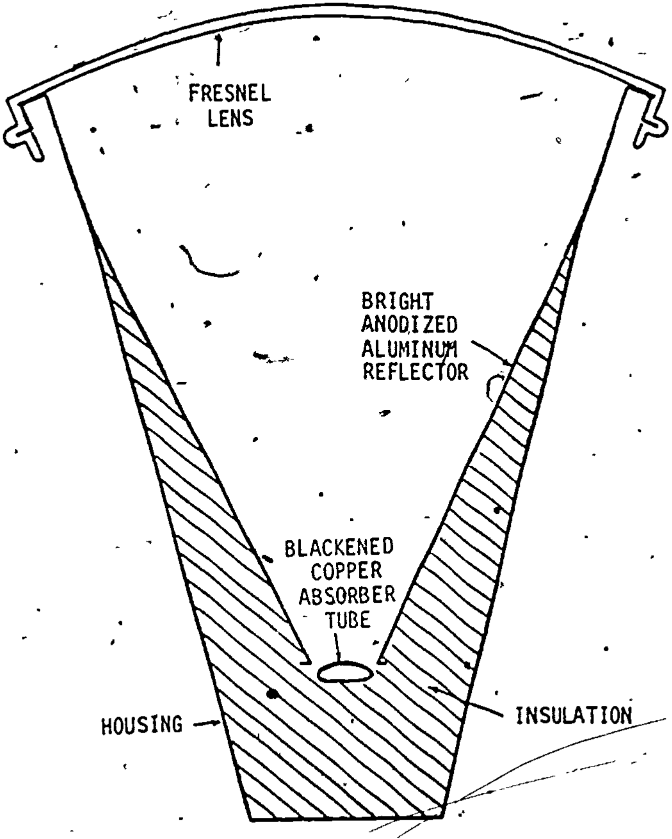
FRESNEL LENS
BRIGHT ANODIZED ALUMINUM REFLECTOR
BLACKENED COPPER ABSORBER TUBE
HOUSING
INSULATION

Figure 22-3. Fresnel Lens Strip Solar Collector.

mounted with the axis in the north-south direction and tilted at an angle with respect to the horizontal plane. The collector rotates from east to west during the day so that the direct rays from the sun are focused into the absorber tube. A distinct disadvantage of concentrating collectors is that only the direct rays from the sun are used, as the diffuse radiation cannot be focused.

THERMAL STORAGE

Considerable research is being devoted toward the utilization of salt hydrates and other phase-change materials for storage of latent heat. The principal difficulties are packaging the storage material and stratification or separation of the material after a few hundred cycles of phase changes. One advantage in the use of phase change materials is supposedly the smaller storage volume required, as compared to water or rocks. However, a solar heating and cooling system requires a water volume of only two gallons of water or one-half cubic foot of rocks per square foot of solar collector area and, in a typical system with 500 square feet of collectors, the water volume needed is about 1000 gallons or about 350 cubic feet of rocks. When packaged phase-change material is arranged in a container with adequate surface contact with the heat transfer fluid from the collectors, it is difficult to achieve a significantly smaller volume of storage.

With proper materials there is, however, an advantage in being able to obtain a sustained constant temperature of the heat delivered from storage. This property of latent heat storage materials can be used to advantage in solar cooling systems, both in the hot storage and cold storage tanks.

Another future prospect for storage of thermal energy is in chemical methods. Chemical storage offers technical possibilities that sensible and latent heat storage do not. These possibilities include; (1) long-term storage without need for insulation and without thermal loss, (2) storage at high energy density, and (3) recovery of stored thermal energy at temperatures above or below the original temperature. Although no thermo-chemical system appears imminent, in concept at least, this method of storage can have important applications in terms of supply and demand and improving thermal efficiency.

HEAT EXCHANGER

The disadvantage of a heat exchanger in present liquid-heating solar systems is the temperature difference needed to transfer the heat at the heat exchanger. A temperature difference of 10 to 20°F has a significant influence on the amount of useful heat delivered by the system. The temperature in storage is low and the collector efficiency is less.

A heat exchanger-storage combination unit is under investigation where heat is transferred from liquid droplets that transport heat from the collector to water in the storage tank. A liquid that is immiscible in water is pumped through the solar collector and through the storage tank as droplets. If the density of the liquid is substantially different from that of water, the liquid droplets will either rise or descend through the water in the storage tank. A schematic of a heat exchanger-storage unit is shown in Figure 22-4. For the illustration shown, the

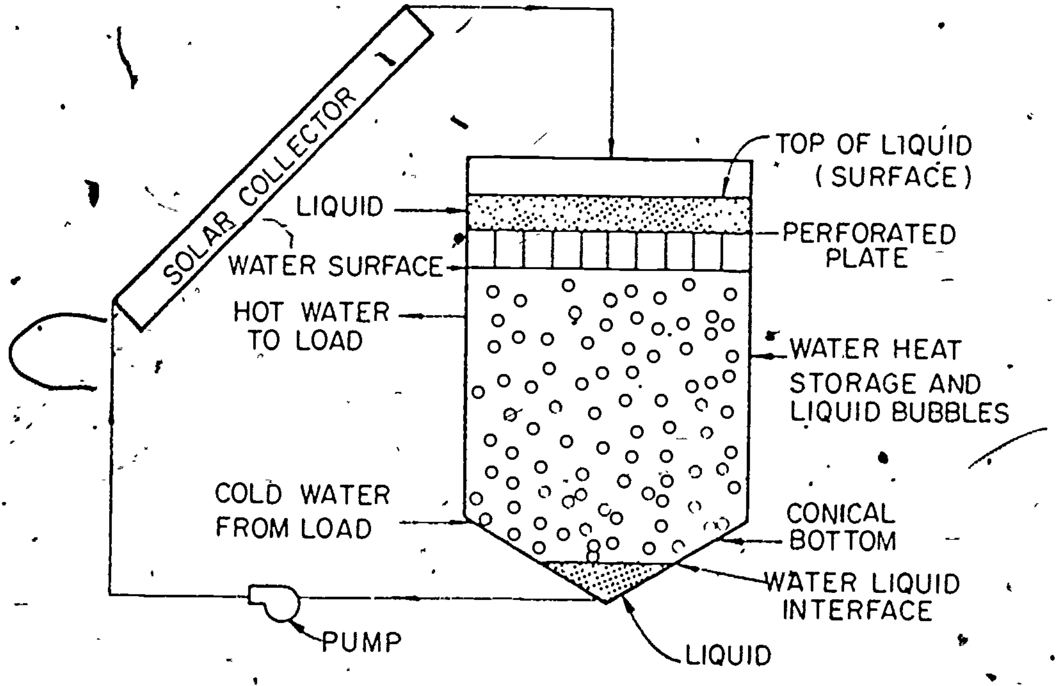
SOLAR COLLECTOR
LIQUID
WATER SURFACE
HOT WATER TO LOAD
COLD WATER FROM LOAD
PUMP
TOP OF LIQUID (SURFACE)
PERFORATED PLATE
WATER HEAT STORAGE AND LIQUID BUBBLES
CONICAL BOTTOM
WATER LIQUID INTERFACE
LIQUID

Figure 22-4. Direct Contact Liquid-Liquid Heat Exchanger

liquid is heavier than water. The liquid is delivered to the top of the tank, is broken up into droplets at the perforated plate, and collects in the bottom cone. The temperature difference between the droplets and the storage water is only about 1°F or less, with substantial heat transfer occurring across the large collective area of the droplets. There are several possible liquids that can be used and, although not named, their properties and approximate costs are listed in Table 22-2.

Table 22-2
Properties of Possible Collector Fluids

Fluid	Freezing Point (°F)	Boiling Point (°F)	Specific Gravity	Cost ($/gal)
1	-31	698	1.116	2.98
2	-36	734	1.208	6.91
3	-31	644	1.048	3.32
4	-41	568	1.120	3.46
5	-27	415	1.043	10.45
6	-13	770	1.162	8.63
7	-76	782	0.927	3.79
8	-67	478	0.913	9.80

SYSTEMS

At present the only commercially available cooling unit in small size that is operable with solar energy is a lithium-bromide absorption cooling unit. As mentioned elsewhere in this manual, there are a number of different experimental cooling units that are being developed, such as the heat engine driven refrigeration machine and ammonia-water continuous-cycle unit.

There is also significant effort being made in the development of so-called total energy systems, where high temperature heat from solar energy is used to generate electricity and the low temperature "waste" heat is used to heat and cool a cluster of buildings. Such systems are likely destined for specialized use in grouped facilities such as military bases but, with some variation, may serve a number of homes or apartment complexes.

In the long term, development of photovoltaic systems for residential buildings is a possibility. Electricity that is generated could operate the heating and cooling system in the house. Whether photovoltaic systems will ever be low enough in cost to be competitive with electricity generated from fossil or nuclear fuels is an open question, but a considerable amount of effort is being devoted to improve efficiency and reduce the costs.

Other improvements in systems which utilize solar energy are hybrid systems consisting of passive as well as active components. There has not been much effort toward development of passive systems except by architectural treatment of windows. While this effort has been significant, more direct heating of residential space with passive systems may minimize the size of the active components and thereby reduce overall costs.

TRAINING COURSE IN
THE PRACTICAL ASPECTS OF
DESIGN OF SOLAR HEATING AND COOLING SYSTEMS
FOR
RESIDENTIAL BUILDINGS

MODULE 23

BUYER'S GUIDE

SOLAR ENERGY APPLICATIONS LABORATORY
COLORADO STATE UNIVERSITY
FORT COLLINS, COLORADO

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INTRODUCTION

In addition to understanding the design and operation of solar heating systems, suppliers and users should be acquainted with several other aspects of solar heating. In order that intelligent selection of equipment can be made, knowledge of industry standards, equipment warranties, performance evaluation data, and related topics is necessary. If evaluations have been performed, their results need to be available to the supplier and user. The kinds of data required for such appraisal must be understood. The advantages and the disadvantages of the main system types for a specific application are particularly important. Knowledge of the type of hardware available, their cost, and their compatibility with other components in the system is essential. Such items as safety and durability are additional criteria for equipment evaluation and selection.

Within this module, the main points enumerated above are addressed, and a guide to their consideration is presented. Because of (a) the newness of the solar equipment industry, (b) limited experience in the use of fully commercial systems in non-subsidized installations, (c) lack of criteria for system evaluation and certification, and (d) lack of information on durability, marketability, and other factors, much of the material here outlined is tentative, rapidly changing, and highly variable in time and place. The following information should therefore be considered a guide rather than a set of specifications.

OBJECTIVE

The objective of this module is to provide the trainee with guides to the purchase of equipment for solar heating systems. The reference list of manufacturers of equipment is not intended to be all inclusive. Guidelines for choosing solar equipment and systems are provided, not only in this module, but throughout this manual.

AVAILABILITY OF SYSTEMS AND COMPONENTS

COLLECTORS

A directory of manufacturers and suppliers of solar heating (and cooling) equipment has been published by the U.S. Energy Research and Development Administration under the title, "Catalog on Solar Heating and Cooling Products". Published in November 1975, and designated ERDA-75, it has been updated by the Solar Energy Industries Association. Among scores of organizations listed as manufacturers of solar heating equipment, possibly a dozen firms have supplied or could furnish solar collectors in quantities of thousands of square feet with one- to two-month lead time for delivery. A listing of some firms is shown in Table 23-1. The list is not intended to be complete nor is the inclusion of a firm intended to imply relative usefulness (efficiency, durability, cost, etc.) of the product. The list contains, however, most of the firms having sold collectors, for space heating, to residential users and to the federal government in total quantities of thousands of square feet. The type of collector manufactured and miscellaneous comments are also presented.

Table 23-1

Selected Collector Manufacturers

Name of Firm	Collector Type	Collector Materials
Ametek	Liquid	Copper, glass (1) or (2)
Chamberlain	Liquid	Steel, glass (2)
General Electric	Liquid	Aluminum, lexan (2)
Grumman	Liquid	Copper, glass (2)
Honeywell	Liquid	Copper-steel, glass (2)
Lennox	Liquid	See Honeywell
Owens-Illinois	Liquid	Glass (evacuated tube)
PPG	Liquid	Copper, glass (2)
Revere	Liquid	Copper, glass (2) or (1)
Solaron	Air	Steel, glass (2)
Sunsource	Liquid	
Sunworks	Liquid or air	Copper, glass (1)

CONTROL

In addition to the equipment listed above, another commercially available component is the control system. The special unit in most solar heating control systems is the differential thermostat with its temperature sensors for insertion in collector and storage. Also available are control panels for connection of the differential thermostat, the room thermostat, and the various relays and motor actuators for blowers, pumps, and valves and dampers. The controllers may be of the conventional electromechanical type with bimetallic temperature sensors or thermocouples or thermistors, along with mechanical relays for energizing motors. Also available are solid-state controllers with thermistor and thermocouple inputs and solid-state switches and relays producing appropriate electric outputs to motors. Electromechanical types are more familiar to heating system installers and service personnel, whereas solid-state units will probably emerge as the more compact and economical system.

Suppliers of control components and special control systems for solar heating include long-established firms in the general control business as well as new companies and groups specializing in specific solar control equipment. A representative list of companies offering differential temperature controllers and complete solar control systems is shown in Table 23-2.

Table 23-2

Selected Suppliers of Solar Heating Controls

Barber Coleman
Deko Labs
Heliotrope General
Honeywell
Penn Controls
Rho Sigma
Robertshaw Controls Company
Solar Controls
(formerly Zia Associates)

HEAT STORAGE

Another important component of the solar heating system is the heat storage unit, but there appears to be no commercial offering of that item. In the liquid system, a conventional tank of some type is purchased. With the air system, a bin is usually constructed on site by the contractor and filled at a suitable time with screened gravel.

COMPLETE SYSTEMS

Several collector manufacturers also provide complete solar heating systems. Their products consist of collectors, accessory hardware for collector support and connection, pumps and/or blowers, preassembled fluid handlers comprising motors, blowers, automatic dampers, filters, water heating coils (for the air system), and motors, pumps, automatic valves (for the liquid system), and controls, including sensors and circuitry for actuating the various motors in the system. Some companies also supply water heating accessories, including heat exchanger and tanks, when that option is involved. The suppliers of complete solar heating systems do not usually furnish a heat storage unit, because its size and local availability usually make its local procurement more practical. Sizing, layout, and detailed design are also offered by some system suppliers. These firms provide the information necessary for installation of their equipment by heating and plumbing contractors having little or no experience in solar equipment installations. Table 23-3 lists a few of the known suppliers of complete solar heating systems.

Table 23-3
Selected Solar Heating System Suppliers

Name of Firm	Type of System
Daystar	Nonfreezing liquid collection and storage
General Electric	Nonfreezing liquid collection and storage
Honeywell	Nonfreezing liquid collection and storage
Piper Hydro	Water collection (nondraining) and storage
Reynolds	Water collection (drainable) and storage
Solaron	Air collection, pebble-bed storage
Solar Utilities Co.	Water collection (nondraining) and storage

EQUIPMENT PERFORMANCE DATA

Most of the suppliers of solar heating system components provide technical data on their performance. Most of the collector data sheets contain information on solar heat collection efficiency at various temperatures and radiation levels. Some include information and instructions for sizing solar heating systems and installation procedures. At least one firm offers an extensive manual covering its products, instructions on their selection and sizing, and their assembly, installation, and servicing.

It should be recognized that some of the manufacturers' literature contains information which has not been verified by impartial analysis, and that the data may not be representative of performance under typical operating conditions. The user is advised to proceed with caution in applying manufacturers' performance figures that have not been independently verified.

Standardized procedures and instrumentation for testing solar equipment have been developed by the National Bureau of Standards (NBS) and are described in two reports:

1. "Method of Testing for Rating Solar Collectors Based on Thermal Performance", NBSIR-74-635. Hill and Kusuda, Center for Building Technology, NBS, December 1974, Interim report prepared for the National Science Foundation.
2. "Method of Testing for Rating Thermal Storage Devices Based on Thermal Performance", NBSIR-74-634. Kelly and Hill, Center for Building Technology, NBS, March 1975, Interim report prepared for the Energy Research and Development Administration.

Although the testing procedures described in these reports are not mandatory for the rating of equipment, they are being accepted by governmental purchasers of solar equipment.

Numerous solar collectors of the liquid heating type have been tested independently by the NASA-Lewis Research Center in Cleveland. Reports of their performance over a range of conditions are available and can be used as a guide to equipment selection. These test results may also be compared with the performance claimed by the manufacturers in their data sheets. Additional testing of liquid heating collectors is also in progress in several independent laboratories.

There have been no independent evaluations and tests of solar air heaters, but facilities are being established at the National Bureau of Standards and at the NASA-Marshall Test Center in Huntsville, Alabama.

Facilities for testing and evaluation of complete solar heating systems are extremely limited. Colorado State University has three identical residential-type buildings in which various systems are being developed and evaluated. This program is producing information which can guide the choice of general system type, and will also yield detailed operating data on specific systems.

SELECTION OF COMPONENTS AND SYSTEMS

Choice of equipment for solar heating involves a knowledge of the characteristics that are significant (and critical) and the advantages and disadvantages of each system type. Besides the information contained in this manual, reference may be made to a helpful government publication, "Buying Solar", published by the Federal Energy Administration, June 1976.

Among the factors most important in equipment choice are the quality of materials and workmanship in the collector, controls, and fluid-handling equipment, the suitability of the materials and equipment to the application (involving such factors as durability, dependability, and safety), heat recovery efficiency over the range of operating conditions encountered, equipment cost, and installation cost.

SELECTION OF SYSTEMS

The system types requiring choice are primarily the flat-plate liquid-heating collector and associated equipment, and the flat-plate air-heating collector with its pebble-bed storage and air handling facility. Another possible choice is a system incorporating an evacuated glass tubular collector in either an air heating or water heating system. So-called passive systems involving collection and storage of heat by materials on or in roofs and walls of buildings rarely are candidates for selection because (a) their practicality has not been proven, (b) there is no manufacturer of such equipment, and (c) if used, these systems are essentially part of the building rather than a heating system. Finally, a system based on use of a focusing collector, although one is commercially available, would seldom be a candidate for residential use because of high cost, tracking requirements, and maintenance demands. Even for commercial buildings, the high cost is a deterrent to general use.

QUALITY OF MATERIALS AND WORKMANSHIP

Durable materials and high-quality workmanship are necessary for efficient, trouble-free operation of solar-heating systems. Visual inspection will often separate the good and poor equipment. Other criteria are records of satisfactory use in previous installations, compliance with minimum property standards, and recommendations from impartial specialists. With liquid systems, the collector, storage unit, heat exchangers, if used, and pumps and piping should be made of materials which are completely compatible with the liquids being used in order that corrosion will not prematurely damage or destroy the system or its components. The collector and other parts of the system must also be able to withstand the maximum and minimum temperatures to which they are exposed. The absorber plate in an efficient collector of the flat-plate type can reach temperatures above 350°F when fluid circulation is interrupted accidentally or purposely, and there should be no material in the collector not capable of withstanding no-flow temperatures for prolonged periods. Wood or other materials which can outgas at these temperatures should never be used in a solar collector. If inspection shows the presence of such materials, the collector is clearly unsuited to normal space heating applications.

SELECTION OF COLLECTOR

The efficiency of the collector in recovering solar energy in a heated fluid is the primary determinant of the size of collector required for supply of a particular fraction of the total heat requirements of a building. And, although this is an important criterion for collection selection, installed cost per unit area is equally significant. Assuming two styles of collectors have equal durability, the one having the greater heat delivery per dollar of first cost is the superior choice, regardless of the efficiency and the cost themselves. In other words, an increase of a few percentage points in efficiency which might be achieved by doubling the cost per square foot is not advantageous. The purchaser should, therefore, base the choice among various collectors of the general type selected on reliable efficiency measurements, delivered price of the collectors, and the cost of installation determined by the installer's bid or the cost of installing similar systems in other buildings. Unless the solar collection efficiency claimed by the manufacturer has been independently verified or reliably confirmed by theoretical analysis, it should not be accepted without question.

As noted in Module 4, the sizing of a solar collector and associated equipment for carrying a certain fraction of the total heating load cannot be based on some collector efficiency measurement at "ideal" conditions characterized by a full sun nearly perpendicular to the collector and at small to moderate temperature difference between collector fluid and the surrounding atmosphere. Seldom is the collector operating at such favorable conditions in normal use, so average efficiencies are far below such a level. In the selection of solar equipment, however, performance of collectors among a single general type can be compared at the ideal conditions. If collector efficiency is reported over a range of solar intensities and temperature conditions, comparison can be made at poor operating conditions as well as the better ones.

The two items probably most commonly overlooked in the selection of solar collectors and other system components are the durability, or apparent useful life, of the equipment and the cost of its installation in the building. The annual cost of ownership of the equipment is approximately inversely proportional to the useful life. In other words, if a solar collector must be replaced in 15 years, there is no advantage in its purchase at half the price of another collector having a 30-year life. Numerous collectors are on the market today which cannot be expected to operate satisfactorily even for 10 years, so their purchase at prices as low as $5 per square foot appears unwise. A collector which costs $12 to $15 per square foot that can be expected to function satisfactorily over the entire life of the building is a far better investment.

COMPARISON OF SYSTEM TYPES

The two major types of systems now available commercially are those which employ a liquid for transfer of heat from collector to storage and those which utilize air for the same purpose. The so-called passive types, in which collection and storage are combined, are not commercially manufactured because they are so closely associated with the design and construction of the building that they are primarily architectural considerations.

Nearly all of the air and water system types involve collectors employing flat-metal absorber plates overlaid with flat-glass sheets. A modification of this design is applied in the several variations of the evacuated tubular collector for air or water heating. A focusing type of collector employing a transparent plastic Fresnel lens is also receiving specialized experimental use.

ADVANTAGES OF LIQUID SYSTEMS

In comparing air and liquid handling in systems, each has advantages and disadvantages. The primary advantages of the liquid system are due to use of a low-cost fluid with high heat capacity. Relatively small piping for transferring heat from collector to storage and from storage to the heated space in hydronic distribution systems is an economic advantage, particularly in large buildings. The volume of water in which a given quantity of heat can be stored is much less than required of any other material not undergoing a phase change of some type. Heat storage in materials undergoing phase changes is not commercially practical, so water is the most compact heat storage material now available.

Another advantage of the liquid system is its capability for solar air conditioning. Although such systems are not fully developed, they do have practical possibilities, particularly in larger industrial and commercial buildings. An additional advantage in the liquid system is the number of commercial manufacturers of liquid heating solar collectors. Various styles, materials (aluminum, copper, and steel), transparent coverings (glass, plastic films, and heavy plastics), and sizes are available. Finally, a large amount of experience is available with liquid collectors (originally used for hot-water supply), including theory as well as practice.

DISADVANTAGES OF LIQUID SYSTEMS

The disadvantages of liquid systems result primarily from the chemical and physical properties of water. Its freezing point, boiling point, and chemical reactivity with metals require designs and materials which can add substantial cost to a solar heating system. In nearly all parts of the United States, water would occasionally freeze in a solar collector and cause extensive damage. A fail-safe drainage system must, therefore, be provided if water is used in the collector, or a non-freezing liquid must be used, with heat exchange to water storage in a part of the building where freezing cannot occur. A self-draining collector imposes some design restrictions, and the periodic filling of the collector tubes with air imposes limitations on the types of metal which can be used. Nonaqueous heat transfer liquids may be used in the collector loop, but their practical utility has yet to be adequately demonstrated.

The corrosiveness of water in contact with aluminum or steel, in the presence of air, is a factor which must be considered in the design and use of water-heating solar collectors. Galvanic corrosion (in the presence of other metals) of aluminum in water must be avoided by suitable non-conducting connections in the system. Pitting corrosion of aluminum in the presence of slight metallic impurities as well as dissolved oxygen and impurities in the water may result in early failure of the aluminum tubes, particularly if thin-walled. Breakdown of anti-freeze solutions (ethylene glycol, for example) to acidic compounds can accelerate corrosive attack and must be avoided by suitable preventive maintenance.

Steel is less subject to attack than aluminum, but precautions must nevertheless be taken. The probable life of a steel collector is greater than that of an aluminum collector having the same tube thickness. Periodic draining and filling with air must, however, be avoided. Copper, at least for tubes, appears to be the most durable and dependable material. The only disadvantage is its substantially higher cost. A plate-type copper collector requires an outlay roughly three dollars per square foot in excess of that for aluminum. At the retail level, this difference could be as much as five to six dollars in selling price.

With any of the metals used for water-heating collectors, corrosion inhibitors can be added to the solution (whether freeze-protected or not) thereby substantially extending the life of the equipment. The inhibitor itself, however, must be maintained at suitable concentration by periodically checking and adding when necessary.

Another disadvantage of the water system is the boiling which occurs if circulation is lost during sunny weather. The system must be designed with appropriate vents or relief valves to permit discharge of steam when these failures occur. If the condition persists for several hours, there can be so much loss of fluid that recharge is then necessary. For typical residential and commercial installations, a maintenance man would have to be called, and additional antifreeze agent (if used), corrosion inhibitor, and water would have to be added. These requirements impose costs which must be considered in any comparison of systems.

In a well-designed and maintained liquid system, damage to the building and its contents from liquid leakage should not occur. However, poor maintenance or careless operation can contribute to leakage of the collector fluid or of water from the storage system through one of many joints and connections, or through corrosion sites, and can result in expensive damage. Good preventive maintenance is therefore a primary requirement of satisfactory operation of a liquid system.

ADVANTAGES OF AIR SYSTEMS

The advantages and disadvantages of an air system are essentially the reverse of those associated with a liquid system. Advantages are the absence of problems associated with corrosion, freezing, boiling, fluid replacement, monitoring of fluid composition, and potential damage by system leakage.

DISADVANTAGES OF AIR SYSTEMS

A disadvantage of the air system is the larger volume required for heat storage - approximately three times that for the equivalent heat storage capacity in water. This requirement imposes a need for floor space having a linear dimension approximately 60 percent greater than for a cylindrical storage tank. Equal heat storage can be provided, for example, in an eight-foot cube of pebbles and in a tank of water five feet in diameter and eight feet high. Another air system disadvantage is the size of ductwork between collector and storage. About four square feet needs to be available for two ducts between collector and storage in a typical residential installation. A third disadvantage is the current lack of air conditioning equipment operable with a solar-heated air supply. This situation is not yet a deterrent to air system use, however, because no solar air conditioning system is yet commercial.

Comparison of the advantages and disadvantages of solar heating system types outlined above leads to the conclusion that the air system is superior insofar as durability and freedom from maintenance are concerned. Experience with a limited number of systems bears out this generalization. As to compactness and wide availability of hardware, the liquid system appears to be the better choice. These relative advantages suggest that air systems may predominate in residential installations where maintenance is notoriously neglected, where compactness is often not considered essential, and where durability is important. Liquid systems, on the other hand, may predominate in commercial and industrial installations where maintenance is routinely practiced, where space is frequently at a premium, and where occasional equipment replacement is acceptable if economically desirable.

SYSTEM PERFORMANCE

In terms of system efficiency, or annual heat delivered per unit collector area, the two systems have comparable performance. Several studies have shown that the difference in heat output is small, and that one system may be slightly better under some conditions and the other superior in other situations. The most recent information on two identical adjacent houses shows nearly one-third more heat was supplied by the air system from equal collector areas. But a conservative appraisal is that the two systems have approximately equal heat delivery capability per square foot of collector area. More data are needed before more definitive statements can be made.

COST OF HEAT DELIVERED

The final and conclusive basis for comparison is cost per unit heat delivered. If efficiency, useful life, and maintenance costs are equal, the system requiring the least maintenance per square foot of collector is the best choice. System costs are not yet sufficiently established for positive selection on this basis. However, examination of published prices of solar collectors and consideration of the costs of other components in the system suggest that the total installed cost of the air system is lower than that of the liquid system, for equal heat output. Evidence in support of this indication is not conclusive, however, so unless actual quotations can be compared, it should be assumed that the cost difference is not large, possibly not over 10 percent of the total investment, and that any difference is probably in favor of the air system.

Another important factor bearing on solar heat cost is the useful life of the system and the costs of maintenance and repairs. On these points there is little doubt that the air system involves lower annual expense. The absence of corrosion, the use of moderate-priced metal (mild steel), and the absence of servicing requirements indicate that the air system will have a longer life and lower maintenance cost than the liquid system.

With respect to evacuated tubular collectors, their high efficiency is a great advantage. These units are not yet being made for general sale, so it is difficult to make comparisons with flat-plate systems. Manufacturing costs are much higher, and current prices may not reflect true costs. But if these units can be produced in large volume (e.g., a thousand tons of glass per month), costs might reach a competitive level. Selection of evacuated tubular systems today would have to be based on criteria other than cost, such as high temperature delivery of collector fluid at reasonable efficiencies. But when demand reaches the level justifying automated tubular collector production with a furnace used exclusively for this product, costs may become very attractive.

There is also a focusing collector (Fresnel lens) which has received some experimental use. It requires a tracking mechanism and the cost is substantially higher than the other systems. Unless high temperatures, well above 200°F, were a specific requirement as, for example, for absorption air conditioning, there appears to be no advantage in the use of this low-concentration focusing system. The considerably higher cost, inability to focus diffuse radiation, and the need for moving hardware, plus maintenance, appear to preclude its practical use for space heating.

In the final choice of a solar heating system, consideration must be given to the type of use which the system is to meet. As previously indicated, liquid systems appear to have some advantages over air systems in large installations where maintenance is customary and where cooling may now or later be provided by solar energy. Other circumstances might also provide incentives for liquid system use. It is evident that both systems have potential for widespread application.

CRITERIA AND STANDARDS

Although no performance criteria or standards for solar heating equipment have been established, several such efforts are being made. Among the active organizations are the American Society for Testing and Materials (ASTM), the American National Standards Institute (ANSI), the American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc. (ASHRAE), the Sheet Metal and Air-Conditioning Contractors' National Association (SMACNA), and various government bureaus, including the National Bureau of Standards (NBS), the Department of Housing and Urban Development (HUD), and the Energy Research and Development Administration (ERDA).

A committee of the ASTM and ANSI organizations is actively engaged in formulating standards for solar heating equipment. No results have been publicly released, but criteria or guidelines may be expected.

ASHRAE, through its series of manuals on heating and air conditioning, continues to expand its section on solar heating and cooling. The 1974 edition of "Applications" contains solar heating information and guidelines in Chapter 59. This material is in the form of a reference handbook for designers and installers of solar heating equipment, but it is comparatively general in its content.

An important project of the National Bureau of Standards is the formulation of performance criteria which solar heating and cooling equipment should be expected to meet. Two of the results of this project are the reports, "Interim Performance Criteria of Commercial and Solar Heating and Combined Heating/Cooling Systems and Facilities", NASA 98M-10001, 28 February 1975 (prepared by NBS) and "Interim Performance Criteria for Solar Heating and Combined Heating/Cooling Systems and Dwellings", HUD, 1 January 1975 (prepared by NBS for HUD). These publications contain information on the characteristics of solar systems and components which are important in the selection of equipment. No requirements are outlined, in terms of quantitative performance, but the equipment is expected to perform at the level which the manufacturer or supplier specifies. In addition to the criteria themselves, the reports describe methods for measuring the performance of collectors and heat storage units.

The next government effort along these lines has resulted in the release of "Intermediate Minimum Property Standards Supplement for Solar Heating and Domestic Hot Water Systems," prepared by the National Bureau of Standards for the Department of Housing and Urban Development (HUD). In conformance with other HUD documents of this type, the specifications outlined are those which solar heating equipment will have to meet if federal funds, such as FHA home loans, are used in financing the structure or its components. As with the "interim performance standards" developed by NBS, the solar heating and cooling standards in the HUD document are directed mainly to safety, durability, reliability, and such factors rather than to the specific efficiency of heat supply or other quantitative criteria. The equipment is required to perform according to the manufacturer's claims.

The work being undertaken by SMACNA is directed toward standards for installation workmanship in solar heating systems. Such factors as the quality of the plumbing, sheetmetal work, and electrical work will be considered.

Standards for testing solar equipment have been the subject of work at the National Bureau of Standards for over two years. A useful report of part of this investigation is "Development of Proposed Standards for Testing Solar Collectors and Thermal Storage Devices", NBS Technical Note 899, issued February 1976.

Another document related to standards and criteria, prepared at the Center for Building Technology of the National Bureau of Standards for the Energy Research and Development Administration, Division of Solar Energy, is "Thermal Data Requirements and Performance Evaluation Procedures for the National Solar Heating and Cooling Demonstration Program." This manual provides detailed information and directions for measuring and evaluating the performance of solar heating and cooling systems.

WARRANTIES

The types of warranties offered by manufacturers of solar heating equipment vary considerably. At the present time, if a supplier provides any warranty, it is of the "limited" type. Under its terms, the equipment is warranted to be free of defects in materials and workmanship, and that if such defects are found within a certain period of time after initial use, correction or replacement will be made without cost to the user. Most of the suppliers of solar equipment do not currently offer any type of warranty. A few, larger companies involved in solar equipment manufacture are offering one-year limited warranties. One company marketing an air system offers a 10-year limited warranty.

There appear to be no manufacturer's guarantees as to thermal efficiency or heat delivery capability of solar equipment. Although manufacturers are providing that type of information in their sales literature, they are not guaranteeing the performance in the field. To a certain degree, this omission is due to the inability of the manufacturer to control the quality of the installation. In addition, manufacturers supplying only certain components of a system, such as the collector, cannot be assured that the other components in the system are correctly selected or integrated with their own product. Thus, inferior performance might well be due to factors other than those controlled by the collector manufacturer. A performance warranty would thus be difficult to establish and maintain.

Still another problem in providing a meaningful performance warranty is the great variation in climate encountered and the practical difficulty in accurately measuring the output of the installed equipment. Instrumentation is usually not provided, so measurement of performance is likely to be an expensive investigation by an experienced engineer. Disputes, litigation, and other problems would be inevitable.

Practical performance warranties should become available for complete solar heating systems provided by a single manufacturer, assembled and installed by a single responsible individual or firm. The manufacturer could then guarantee the system to the installing firm which, in turn, would guarantee it to the purchaser. In case of dispute, the installer could measure system performance in the presence of the owner and a third party, if demanded, for determination of conformance. If inadequate, corrections would be made in compliance with the warranty, and the installer and manufacturer would establish responsibility for the departure from specifications.

Such developments as the Home Owners Warranty (HOW) program, sponsored by the National Association of Home Builders, can be expected to have an influence on solar heating equipment guarantees. Under the HOW program, all defects in a residential structure will be corrected at no cost to the owner during the first three years of use. It may be expected that solar heating equipment will have warranties conforming with such a program. Manufacturers will then be required to guarantee to the dealer and installer the necessary support for compliance with this program.

The solar equipment manufacturing industry unfortunately includes several small suppliers having practically no experience with solar equipment and offering no warranties of any kind. Purchasers of such equipment have very little chance of reimbursement for costly failures. Even if a small, marginal manufacturer offers some sort of warranty, a purchaser does not have much assurance that the manufacturer will remain in business long enough to make good on its guarantee. In the event of equipment defect or failure, the owner (or installer, if guaranteed by him) would suffer the loss. These and other topics are discussed in the previously mentioned government report, "Buying Solar", published in June 1976 by the Federal Energy Administration and HUD.

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